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6.5 Plant and Animal Husbandry: *Chairman*: R. E. Stewart, F. N. Andrews, T. E. Bond, M. K. Fahnestock, R. L. Givens, G. R. Nelson, H. J. Thompson, R. G. Yeck.

Group 7—Unitary Equipment—L. K. Warrick, Coordinator

7.1 Commercial Food Storage and Dispensing Equipment: *Chairman*: J. H. Rainwater, R. D. Bartlett, D. E. Friedman, L. J. Gibbas, J. R. Haynes, G. W. Mathis, A. Perez, M. H. Strang.

7.2 Domestic Refrigerators and Unit Food Freezers: *Chairman*: P. E. Davey, R. S. Buchanan, H. Deuker, R. L. Eichhorn, W. R. Johnson, Hershel Powell, R. Roider, H. A. Whitesel.

7.3 Truck and Railway Car Refrigeration Units: *Chairman*: G. A. Gallagher, G. E. Anderson, M. B. Green, Paul Jung, H. O. Kirkpatrick, C. W. Phillips, W. H. Redit, James Whalen.

7.4 Drinking Water and Beverage Coolers: *Chairman*: E. W. Scott, L. J. Anderson, W. M. Barnes, F. O. Graham, K. J. Helsing, P. R. Lynn, M. A. Porter, A. L. Rankin, Jr., William Taylor, A. A. Zollo.

7.5 Ice Makers: *Chairman*: D. E. MacLeod, Byron Booher, Crosby Field, A. G. Larson, B. D. Maseritz, Glenn Muffly.

7.6 Air Conditioners and Dehumidifiers: *Chairman*: B. J. Homkes, G. L. Biehn, T. G. Crider, William Hood, R. W. Kelto, H. R. Krueger, F. W. Osborn, D. L. Pitman, W. T. Rouse, L. J. Sahs, L. L. Smith, E. C. Tanner, R. G. Werden, P. W. Wyckoff.

7.7 Railway and Vehicle Air Conditioning: *Chairman*: J. M. Strauss, D. C. Albright, J. V. Dobbs, J. D. Loveley, C. W. McElroy, Meyer Sutton, J. J. Wertheimer.

7.8 Milk Coolers: *Chairman*: R. L. Latzko, Darrell Evans, C. V. Haverly, Jr., R. E. Steinhorst, D. F. Swanson, T. L. Tyler.

Group 8—Products and Processes—E. J. Robertson, Coordinator

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8.2 Dairy, Poultry and Allied Products: *Chairman*: E. N. Kerrigan, T. Y. Davis, L. J. Gibbas, J. Kronholz.

8.3 Fermentation Processes and Allied Products: *Chairman*: B. H. Bishop, W. E. Helin, M. J. Mayer, Fred Ophuls, C. M. Sieben, Charles Torry.

8.4 Concentration, Dehydration, and Other Food Preservation Processes: *Chairman*: D. K. Tressler, D. C. McCoy, J. C. Woodroof.

8.5 Agricultural Products: *Chairman*: D. H. Dewey, C. A. Anderson, S. E. Mapes, W. E. Martin, C. Matthiesen, A. J. Monta.

8.6 Frozen Foods: *Chairman*: C. F. Evers, Edgar Dickson, H. C. Diehl, W. H. Redit, M. F. Tokach.

8.7 Ice Making, Skating Rinks, and Industrial Refrigeration: *Chairman*: W. O. Kline, E. A. Johnson, F. W. Knowles, R. A. Stencel, Herman Vetter, O. B. Wert, C. L. Whitaker.

8.8 Cold Storage Warehouses and Locker Plants: *Chairman*: G. A. M. Anderson, H. C. Brown, Jr., B. B. Green, O. K. Irvine, W. E. Martin, E. K. Strahan, R. M. Stern, R. H. Trinkle.

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2nd Vice President.....A. F. Worden, Jr.
Secretary.....R. Jorgensen
Treasurer.....R. L. Jameson
Board of Governors: F. R. Collins, Jr., K. E. Champagne, J. Davis, C. W. Stone, P. D. Wyckoff

Niagara Peninsula

Headquarters, Hamilton, Ont.

President.....W. M. Carr
Vice President.....W. M. McDonald
Secretary.....H. J. Kallman
Treasurer.....F. Caldwell
Board of Governors: D. Harper, D. A. McCoppen, J. A. Mitchell, M. V. Murray

North Jersey

Headquarters, Newark

President.....C. E. Parmelee
1st Vice President.....H. Wolf
2nd Vice President.....C. Zimmer
Secretary.....L. Lieberman
Assistant Secretary.....F. C. Hawco
Treasurer.....H. Fox
Board of Governors: C. Collins, G. V. Dennis, G. Freeman, L. Larkin

North Piedmont

Headquarters, Greensboro, N. C.

President.....R. D. Funderburk
Vice President.....E. H. G. Farthing

Secretary....................S. M. Blount, Jr.
Treasurer....................T. C. French
Board of Governors: D. T. Waynick, J. Swaim, R. F. Bean

Northeastern New York

Headquarters, Albany

President....................B. E. Mullen
1st Vice President....................H. V. Cross
2nd Vice President....................A. Petrecki
Secretary....................C. Batchelder
Treasurer....................P. Enright
Board of Governors: L. M. Brown, F. Sheldon, L. LaPlante

Northeastern Oklahoma

Headquarters, Tulsa

President....................J. Shaw
Vice President....................W. Smith
Secretary....................R. L. Herron
Treasurer....................L. Bury
Board of Governors: J. Tumilty, R. Shoemaker, J. Nichols

Northern Alberta

Headquarters, Edmonton, Alta.

President....................R. Proudfoot
Vice President....................H. Hole
Secretary....................A. Nordstrom
Treasurer....................H. Bristow
Board of Governors: A. Stix, V. Carroll, E. Pantel, H. Fisher, A. McCallum, H. Wiber

Northern Connecticut

Headquarters, Hartford

President....................A. S. Decker
1st Vice President....................E. M. Johnson
2nd Vice President....................H. Cosentino
Secretary....................I. M. Fierberg
Treasurer....................E. A. Nichols, Jr.
Board of Governors: A. S. Decker, D. Allen, H. Cosentino, E. Nichols, I. M. Fierberg, F. Raible, Jr., E. Johnson

Ontario

Headquarters, Toronto, Ont.

President....................E. Fox
1st Vice President....................J. D. Coates
2nd Vice President....................W. A. Mould
Secretary-Treasurer....................J. B. Parker
Board of Governors: G. Granek, F. D. Ledgett, E. J. Okins, H. R. Roth, G. C. Toms, W. R. Woodcock, L. N. Adams

Oregon

Headquarters, Portland

President....................E. S. Constant
Vice President....................J. L. Waymire
Secretary....................C. W. Timmer
Treasurer....................W. D. Maxwell
Board of Governors: O. T. Jacobson, K. G. Stahl, C. J. Bell

Ottawa Valley

Headquarters, Ottawa, Ont.

President....................P. G. Fortier
Vice President....................J. Partington
Secretary....................I. Paterson
Treasurer....................D. McKeen
Board of Governors: J. Bowie, M. Overbury, A. Simmonds

Philadelphia

Headquarters, Philadelphia, Pa.

President....................O. M. Kershock
1st Vice President....................D. S. Plewes
2nd Vice President....................A. A. Lincoln
Recording Secretary....................O. J. Nussbaum
Corresponding Secretary....................K. M. Wicks
Treasurer....................P. R. Anderson
Board of Governors: L. Mack, J. C. Benson

Pittsburgh

Headquarters, Pittsburgh, Pa.

President....................G. E. Smetak
Vice President....................A. S. Estatico
Secretary....................D. B. Hicks
Treasurer....................J. H. Llewellyn
Board of Governors: K. M. Newcum, R. M. Toucep, C. Stanger

Puget Sound

Headquarters, Seattle, Wash.

President....................K. Massart
1st Vice President....................R. R. Kirkwood
2nd Vice President....................D. Hopkins
Secretary....................D. J. Moore
Treasurer....................H. A. Bickel
Board of Governors: M. R. Overbye, D. Ervine, F. Nuyens

Rhode Island

Headquarters, Providence

President....................R. E. Wilkinson
Vice President....................D. J. Kiely
Secretary....................H. Ring
Treasurer....................J. K. MacLean

Board of Governors: L. M. Dunlap, M. Zimmerman, F. T. Allen, L. J. McPherson, F. J. Sarra, J. D. Guillmette

Richmond

Headquarters, Richmond, Va.

President.................M. M. Alley
Vice President.................J. C. Turlington
Secretary.................J. G. Johnson, Jr.
Treasurer.................A. B. Miller
Board of Governors: F. J. Weiss, G. J. Wachter, H. A. Garber, M. O. Roache

Rochester

Headquarters, Rochester, N. Y.

President.................S. J. Stachelek
1st Vice President.................D. A. Sweetland
2nd Vice President.................G. C. Boncke
Secretary.................D. D. Hinman, Jr.
Treasurer.................H. E. Siebert
Board of Governors: J. D. Johnson, B. C. Teal, H. J. Dyminski

Rocky Mountain

Headquarters, Denver, Colo.

President.................J. Reed
1st Vice President.................R. G. Pritchard
2nd Vice President.................L. D. Niblack
Secretary.................J. L. Crellin
Treasurer.................S. Sellers
Board of Governors: L. Bindner, E. Heckman, J. Cipra, Jr.

Sacramento Valley

Headquarters, Sacramento, Calif.

President.................L. S. Stecher
Vice President.................J. R. Handy
Secretary.................M. Laks
Treasurer.................D. Yoshpe
Board of Governors: W. B. Lander, F. H. Taylor W. Nero

St. Louis

Headquarters, St. Louis, Mo.

President.................F. G. Meyers
Vice President.................J. B. Killebrew
Recording Secretary.................H. R. Halt
Corresponding Secretary.................C. F. Barmeier
Treasurer.................C. W. Baker
Board of Governors: K. Williams, C. Hartung, J. Cuba

San Diego

Headquarters, San Diego, Calif.

President.................M. Jackson
Vice President.................W. Neild
Secretary.................C. H. Deilgat
Treasurer.................S. Christian
Board of Governors: S. Bayne, D. M. Butala, K. Klien

San Joaquin

Headquarters, San Joaquin, Calif.

President.................D. Rodkin
Vice President.................W. L. Grotzke
Secretary.................G. Yamaguchi
Treasurer.................R. L. Adams
Board of Governors: R. Cody, W. L. Donley, L. I. Horton

Savannah

Headquarters, Savannah, Ga.

President.................E. F. White
Vice President.................R. E. Kinser
Secretary.................J. H. Baker
Treasurer.................J. M. Cates
Board of Governors: R. Graig, C. Courtney

Shreveport

Headquarters, Shreveport, La.

President.................R. L. Johnson, Jr.
Vice President.................J. Tarlton
Secretary.................F. Adams
Treasurer.................F. H. Spaulding
Board of Governors: W. Jarvis, F. Adams, J. Guth, Q. W. Hargrove

South Carolina

Headquarters, Columbia

President.................W. O. Blackstone
Vice President.................R. F. Lindsay
Secretary.................H. R. King
Treasurer.................J. C. Boles
Board of Governors: T. O. Curlee, B. T. Bootle, J. C. Harrison

South Dakota

Headquarters, Sioux Falls, S. D.

President.................R. Larson
Vice President.................J. Sandfort
Asst. Vice President.................D. Rosenstein
Secretary.................J. Mahanna
Asst. Secretary.................W. Menzel
Treasurer.................F. Monick
Board of Governors: M. Bird

South Florida

Headquarters, Miami

President.......................A. Cowan
Vice President.......................J. E. Beard
Secretary.......................A. R. Dickterenko
Treasurer.......................K. Cunningham
Board of Governors: J. Lotz III, A. R. Martin, Jr., J. L. Middleton

South Piedmont

Headquarters, Charlotte, N. C.

President.......................W. T. Foreman
Vice President.......................D. Rickelton
Secretary.......................W. Bainbridge
Treasurer.......................B. W. Scott, Jr.
Board of Governors: E. V. Quercash, J. M. McDermott, D. P. Schiwetz

Southern Alberta

Headquarters, Calgary, Alta.

President.......................N. J. Howes
Vice President.......................E. W. Deeves
Secretary.......................O. Reggin
Treasurer.......................A. V. Willis
Board of Governors: J. D. Orr, D. C. Bell, J. H. Hole, F. Thompson, W. W. Roe

Southern California

Headquarters, Los Angeles

President.......................J. C. Hall
Vice President.......................B. L. Hutchinson, Jr.
Secretary.......................J. R. Hall
Treasurer.......................V. J. Burke
Board of Governors: A. A. Hellman, M. Kodmur, R. H. Jorgensen, P. A. Van Woerkom

Southern Connecticut

Headquarters, New Haven

President.......................E. Olsson
1st Vice President.......................G. A. Dion
2nd Vice President.......................E. Morley
Secretary.......................S. Bronski
Treasurer.......................D. Harrigan
Board of Governors: G. DeFeo, C. McElroy, R. Cahoon, B. Packtor

Toledo

Headquarters, Toledo, Ohio

President.......................J. S. Meyer
Vice President.......................R. C. Moorhead
Secretary.......................D. W. Duston
Treasurer.......................G. W. Bleckner
Board of Governors: N. W. Dawe, J. E. Wilkie, G. L. Heiser, G. Frost

Tucson

Headquarters, Tucson, Ariz.

President.......................D. D. Shipley
Vice President.......................J. P. Jones
Secretary.......................A. E. Hamilton
Treasurer.......................B. M. Dehlinger
Board of Governors: F. H. Blackmore, D. M. Heskett

Utah

Headquarters, Salt Lake City

President.......................C. E. Warren
Vice President.......................J. N. Eichers
Secretary-Treasurer.......................R. W. Williams, Jr.
Board of Governors: L. Peterson, G. Wilcox, V. Q. Tregeagle, R. Oliver

West Texas

Headquarters, Lubbock

President.......................O. R. Downing
Vice President.......................J. F. Roberts
Secretary.......................H. W. Bartlett
Treasurer.......................T. A. Niblack, Jr.
Board of Governors: W. R. Anthony, Henry Bartlett, T. F. Sartor

Western Massachusetts

Headquarters, Springfield

President.......................J. C. Martin
1st Vice President.......................K. W. Maki
2nd Vice President.......................J. Tropp
Secretary.......................T. Best
Treasurer.......................W. A. Rochford
Board of Governors: F. K. Hurley, J. J. Curran

Western Michigan

Headquarters, Grand Rapids

President.......................G. L. Jepson
1st Vice President.......................D. A. Rackliffe
2nd Vice President.......................W. Wessels
Secretary.......................L. H. Hinkel
Treasurer.......................C. W. DeKorte
Board of Governors: S. Curtis, F. J. Belton

Wichita

Headquarters, Wichita, Kans.

President.......................R. Peugh
1st Vice President.......................P. Bauer
2nd Vice President.......................D. L. Manson
Secretary.......................R. L. Surtees
Treasurer.......................K. Stewart
Board of Governors: R. Peugh, P. Bauer, D. L. Manson, R. L. Surtees, K. S. Stewart, F. Osborn

Wisconsin

Headquarters, Milwaukee

President . . . . . W. H. Miller
1st Vice President . . . . . K. F. Waraczynski
2nd Vice President . . . . . F. W. Goldsmith
Secretary . . . . . G. A. Adlam
Treasurer . . . . . R. H. Schulz
Board of Governors: J. Douglas, J. Eagle, J. Illingworth

Special Branch

Panama and Canal Zone

Headquarters, Curundu, Canal Zone

President . . . . . I. Fong
1st Vice President . . . . . R. A. Bettis
2nd Vice President . . . . . R. B. Mouynes
Secretary . . . . . C. D. Speier
Treasurer . . . . . R. G. Fifer

Overseas Branches

Switzerland

Headquarters, Zurich

President . . . . . A. Kummer
Vice President . . . . . H. Parli
Secretary . . . . . T. Kohli
Treasurer . . . . . M. Koenig
Headquarter . . . . . H. Brown

Italy

Headquarters,

President . . . . . G. F. Bertolini
Vice President . . . . . L. Chieregatti
Secretary . . . . . G. duBot
Treasurer . . . . . U. Stefanutti
Board of Directors: G. F. Bertolini, L. Chieregatti, G. duBot, U. Stefanutti, L. Jouane, A. Stradelli, M. Santagostino, E. Barbieri, L. Mazzini

Student Branches

Auburn University

Headquarters, Auburn, Ala.

Faculty Advisor . . . . . Prof. T. Min

California State Polytechnic College

Headquarters, San Luis Obispo

Faculty Advisor . . . . . Prof. J. A. Hayes

North Carolina State College

Headquarters, Raleigh

Faculty Advisor . . . . . Prof. R. B. Knight

Oregon State College

Headquarters, Corvallis

Faculty Advisor . . . . . Prof. G. E. Thornburgh

Purdue University

Headquarters, W. Lafayette, Ind.

Faculty Advisor . . . . . Prof. F. B. Morse

Texas A. & M. College

Headquarters, College Station

Faculty Advisor . . . . . Prof. L. S. O'Bannon

University of Detroit

Headquarters, Detroit, Mich.

Faculty Advisor . . . . . Prof. J. B. Olivieri

University of Toronto

Headquarters, Toronto, Ont.

Faculty Advisor . . . . . Prof. F. G. Ewens

BY-LAWS

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS

(Incorporating those amendments approved by the membership at the 67th Annual Meeting at Vancouver, the Semiannual Meeting in Chicago and the 68th Annual Meeting in Denver)

ARTICLE I—Incorporation

Section 1.1 This is a corporation duly organized and existing under and pursuant to the Membership Corporations Law of the State of New York and is hereafter called the "Society."

Section 1.2 The object of the Society is to advance the arts and sciences of Heating, Refrigeration, Air Conditioning, and Ventilation, and the allied arts and sciences, for the benefit of the general public.

ARTICLE II—Government

Section 2.1 The Society shall be governed by the Laws of the State of New York, its Certificate of Consolidation, its By-laws, the Rules promulgated by the Board of Directors in harmony therewith, and all amendments to the foregoing.

Section 2.2 The Society shall neither approve any engineering project or commercial product, nor allow its imprint or name to be used in any commercial work or business, except that it shall be permissible for a manufacturer to state in any manner deemed proper that a product has been tested, or tested and rated, in accordance with an ASHRAE Standard, giving the number of the Standard, however, the Society shall not engage in the testing or rating of such products in behalf of any manufacturer.

Section 2.3 Matters pertaining to politics, religion, or solely to trade shall not be discussed at any meeting of the Society, nor be included in any of its publications.

Section 2.4 As used in these By-laws, the use of "Member" (with a capital M) refers to a member of that grade or higher as set forth in *Section* 3.1, and the use of "member" (with a small m) refers to a member of any grade.

Section 2.5 Official stationery of the Society shall be used only for its official business, and only by its officers, directors, committee members, Chapter and Branch officers, Chapter and Branch committee members, and members of its staff.

Section 2.6 The administrative year of the Society shall begin at the Annual Meeting upon the installation of the new officers and terminate at the following Annual Meeting upon the installation of their successors. The fiscal year shall be determined by the Board of Directors.

Section 2.7 All business meetings of the Society, Board of Directors, Executive and other Committees shall be governed by the rules of procedure contained in Roberts Rules of Order, Revised, where the same are not inconsistent with the law or the provisions of the Certificate of Consolidation, the By-laws, or special rules of order of the Board of Directors.

Section 2.8 The Board of Directors shall designate one or more of the publications of the Society for the publishing of official notices to the members.

ARTICLE III—Membership

Section 3.1 The grades of membership in the Society shall be designated as follows: (A) Honorary Members, (B) Presidential Members, (C) Fellows, (D) Life Members, (E) Members, (F) Associate Members, (G) Affiliates, and (H) Students.

Section 3.2 Honorary Member—Any notable person of preeminent professional distinction may be elected an Honorary Member.

Section 3.3 Presidential Member—Upon the installation of his successor, the outgoing President of the Society shall become a Presidential Member. Past Presidents of either predecessor society shall become Presidential Members.

Section 3.4 Fellow—A Member who has attained unusual distinction in the arts relating to the sciences of heating, refrigeration, air conditioning or ventilation, or the allied arts and sciences, or in the teaching of major courses in said arts and sciences, or who by reason

of invention, research, original work, or as an engineering executive on projects of unusual or important scope, has made substantial contribution to said arts and sciences, and who has attained the age of forty-five (45) years, and has been in good standing as a Member for a period of at least ten (10) years prior to the date of his proposal for Fellow grade.

Section 3.5 Life Member—A Member who has been in good standing for thirty (30) years; and who has attained the age of sixty-five (65) years. He shall retain all the rights and privileges of his former membership grade. He shall not be required to pay any annual dues or any other fees.

Section 3.6 A Member, at the time of his admission or advancement:

(*A*) Shall be a graduate of an engineering curriculum accredited by the Engineers Council for Professional Development and approved by the Board of Directors, or approved by the Board of Directors, and shall have had no less than six (6) years active practice in the professions of engineering or teaching, or both, of which five (5) years shall have been in responsible charge of such teaching or engineering work, and who is qualified to direct such work or carry on important research or design in the engineering field; or,

(*B*) If not such a graduate, shall have equivalent attainments including at least twelve (12) years active practice in the professions of engineering or teaching, or both, of which five (5) years shall be in responsible charge of such teaching or engineering work, all of a character satisfactory to the Board of Directors.

A license to practice Professional Engineering issued by a legally authorized body whose requirements as to education and active practice are considered satisfactory and adequate by the Board of Directors may be considered equivalent to fifty (50) percent of the active practice requirements.

Section 3.7 An Associate Member shall at the time of his admission or advancement to the grade, be a graduate of an engineering curriculum accredited by the Engineers Council for Professional Development and approved by the Board of Directors, or approved by the Board of Directors. If not such a graduate, he shall have equivalent attainments including at least eight (8) years of engineering experience, all of a character satisfactory to the Board of Directors.

Section 3.8 An Affiliate shall have had experience in technical matters, design, operation, or maintenance in heating, refrigerating, air conditioning or ventilating fields, or shall have an interest in the advancement of the Society's aims, and shall possess sufficient qualifications to co-operate with heating, refrigerating, air conditioning or ventilating engineers in the advancement of the knowledge relating to heating, refrigerating, air conditioning, or ventilating engineering and their application.

Section 3.9 A Student shall be a person matriculated in a degree-granting or graduate school with a curriculum accredited by the Engineers Council for Professional Development or the Engineering Institute of Canada, or in a school with a curriculum recommended by the Education Committee and approved by the Board of Directors and pursuing a course of study in preparation for the engineering profession. The Student status shall terminate one year after graduation from the school, or one year from the time he leaves the school.

Section 3.10 The voting membership shall consist of Honorary Members, Presidential Members, Life Members, Fellows, Members, and Associate Members.
All members thus entitled to vote may be called herein "voting members."
Affiliates and Students shall have no right to vote.

Section 3.11 No member shall describe himself in connection with the Society in any advertisement, letterhead, printed matter, or any other manner other than as an Honorary Member, Presidential Member, Life Member, Fellow, Member, Associate Member, Affiliate, or Student, as the case may be, except in official business of the Society.

Section 3.12 The rights and privileges of a member shall be personal to himself and shall not be delegated or transferred, except that each member entitled to vote may vote in person or by written proxy given to another member entitled to vote and dated within three months, which proxy shall be subject to the provisions as set forth in *Section* 6.1

Section 3.13 All right, title and interest of a member in the Society, or its property, shall cease on the termination of his membership by death, resignation or otherwise, and shall vest in the Society.

Section 3.14 Each member, upon his election to membership, shall thereupon be bound by the provisions of the Certificate of Consolidation, By-laws, and rules of the Board of Directors, and all amendments thereto.

Section 3.15 The emblem of membership in the Society shall be worn only by members in good standing.

Section 3.16 Any member may resign at any time by his written request received by the Executive Secretary, provided his dues are paid in full.

Section 3.17 If any Fellow, Member, Associate Member, or Affiliate shall fail to pay his dues by Oct. 1, he shall be classed as delinquent, and if a voting member shall lose his right to vote; if such dues are not paid by Jan. 1, he shall be classed as not in good standing and his membership shall be suspended; if such dues are not paid by June 1, the Executive Secretary shall notify the suspended member by registered mail that unless such dues are paid by June 30, he shall cease to be a member of the Society, and upon his failure to cure such default by June 30, his membership in the Society shall cease; if any Student shall fail to pay his dues by June 30, the delinquent Student's membership shall cease and the Executive Secretary shall notify such Student by registered mail that his membership in the Society has ceased; provided that upon written application satisfactorily explaining a default, accompanied by payment of dues, the Board of Directors may, in its discretion, rescind any forfeiture of membership.

Section 3.18 A former member who has resigned, or who has been dropped from membership, may be reinstated by payment of the same fees charged a new member, or may be reinstated as of his original date of membership if he pays all dues which would have accrued.

Section 3.19 The Board of Directors may, by a two-thirds vote of all the members thereof, censure, suspend or expel any member for misconduct in his relations to the Society, after written preferment of charges, thirty (30) days' written notice of hearing sent by registered mail, and an adequate opportunity to be heard before the Board of Directors or a committee of one or more Members designated by the Board of Directors.

Section 3.20 All applicants for admission to the Society, or for advancement in grade of membership except such as are conferred as an honor, shall make application in such form and with such information as shall be required by the Board of Directors.

Section 3.21 Membership in the Society and advancement in grade shall be in accord with the procedures and the conditions set forth below.

Section 3.22 All applications for admission to the Society in all grades except Student grade and all applications for advancement in membership grade shall be referred to the Admissions and Advancement Committee for investigation and report to the Board of Directors with recommendation as to grade. As soon as practicable after the report, the Board of Directors shall act upon each application by letter ballot. Two (2) disapprovals shall reject any applicant. One (1) disapproval shall require the resubmission of the application to the Admissions and Advancement Committee for further study and it shall make recommendation to the Board of Directors and the Board of Directors shall vote upon the application at its next regular meeting.

Section 3.23 Application for admission to the Society in the Student grade of membership properly endorsed by a member of the engineering faculty of the degree-granting or graduate school stipulated in *Section* 3.9 shall be referred to the Admissions and Advancement Committee for acceptance. Upon approval of the application by the Admissions and Advancement Committee, the Executive Secretary shall be empowered to issue Student memberships to the approved applicants.

Section 3.24 Before submission to the Admissions and Advancement Committee of an application for election as, or advancement to, Affiliate, Associate Member or Member, the name of the applicant shall be published in an issue of the Society's official journal.

Section 3.25 The grade of Honorary Member shall be conferred on no more than three (3) persons in any calendar year. The grade of Fellow shall be conferred on no more than fifteen (15) Members in any calendar year.

Section 3.26 Nominations for Fellow shall be made by the Honors and Awards Committee to the Board of Directors or by petition of not less than ten (10) Fellows and Members. Election shall be by the Board of Directors by secret ballot and more than two (2) negative votes shall defeat the proposal.

Section 3.27 Nominations for Honorary Member shall be made by the Honors and Awards Committee to the Board of Directors. Election shall be by the Board of Directors upon unanimous vote by secret ballot.

ARTICLE IV—Fees and Annual Dues

Section 4.1 Initiation and advancement fees and annual dues shall be fixed by the Board of Directors from time to time and shall be payable as determined by the Board of Directors, and shall be published periodically in the official publication of the Society.

Section 4.2 The Board of Directors may provide for a paid-up membership by one payment of a sum fixed by the Board of Directors and such member retain for life all the rights and privileges of his membership grade before such payment was made, unless otherwise deprived thereof.

Section 4.3 Any member who paid dues for thirty (30) years shall be permanently exempt from the payment of further dues, and shall retain all the rights and privileges of his membership grade, unless otherwise deprived thereof.

Section 4.4 The Board of Directors, for good and sufficient reasons, may waive the dues of any member and the initiation fee of any former member applying for reinstatement who at the time of his resignation was in good standing.

Section 4.5 Members of all grades shall receive the official journal of the Society. Members of all grades, except Student and dues paying members whose pro-rated dues amount to less than 50 per cent of a Member's dues, shall be entitled to receive the Society's other publications as authorized by the Board of Directors.

ARTICLE V—Board of Directors and Officers

Section 5.1 The Board of Directors shall consist of the President, the 1st Vice President, the 2nd Vice President, the Treasurer, the immediate Past President; nine (9) Directors-at-Large, with equitable distribution from the three (3) major areas of membership interest, namely, (1) heating, (2) refrigeration, and (3) air conditioning and ventilation; and ten (10) Regional Directors, each from his respective Region. The nine (9) Directors-at-Large shall be divided into three (3) groups of three (3) each in each area of membership interest, and the members in each group shall hold office for three (3) years, and until their successors shall have been elected and installed. Three (3) Directors-at-Large shall be elected at each Annual Meeting, and also such additional number, if any, as may be necessary to fill vacancies.

The ten (10) Regional Directors shall be elected for a staggered term of three (3) years; a 3-3-4 sequence of the group of ten (10) to be elected annually at the Annual Meeting.

Section 5.2 The Board of Directors shall hold regular meetings at approximately the time of the Semi-Annual and Annual Meetings of the Society.

Special Meetings of the Board of Directors may be called by the President or by three (3) members of said Board. The Board of Directors shall keep a record of its proceedings and shall report on its activities at each meeting of the Society and at the Annual Meeting shall present a written report as required by the Membership Corporations Law of the State of New York.

A quorum of the Board of Directors shall consist of thirteen (13) of its members.

Section 5.3 The Board of Directors shall have full and complete management and control of the activities, properties, and funds of the Society, subject to the provisions of law, the Certificate of Consolidation, and the By-laws.

The Board of Directors may, in its discretion, refer to the Society any important question pertaining to the Society, and shall refer any such questions to the Society upon a majority vote taken at a stated or Special Meeting of the Society.

Section 5.4 The Officers of the Society shall be President, 1st Vice President, 2nd Vice President, Treasurer, and Executive Secretary. The President shall not be eligible for immediate re-election to that office. Elected officers shall receive no salary, emolument or compensation for services rendered to the Society, and shall serve for one (1) Society year or until their respective successors shall be elected and installed.

Section 5.5 All officers of the Society shall perform the duties customarily attaching to their respective offices under the laws of the State of New York, and such other duties and services incident to their respective offices as are delegated to them in these By-laws and as may from time to time be assigned to them by the Board of Directors.

Section 5.6 The terms of all Officers and Directors shall commence upon their installation during the Annual Meeting and shall continue until their successors have been elected and installed.

Section 5.7 Only Honorary Members, Presidential Members, Life Members, Fellows and Members shall be eligible for election as President, Vice President, Treasurer, or Director.

Section 5.8 At all meetings of the Society and of the Board of Directors, the President, or in his absence the Vice Presidents in order of seniority, or in their absence the Treasurer, or a Member selected by the Board of Directors, shall preside.

A vacancy in the office of President shall be filled by the ranking Vice President, or in the event of vacancies in the offices of both Vice Presidents, by election by the Board of Directors. In the event of a vacancy in any other office or directorship the same shall be filled by election by the Board of Directors until the next Annual Meeting.

Section 5.9 The Treasurer shall have custody of the funds of the Society and the Society's books of account, which shall be opened to the inspection of any member of the Board of Directors.

Section 5.10 The expenditures of the Society's funds shall be governed by the Budget as approved, modified, or from time to time amended, by the Board of Directors, and no additional expenditures shall be made without the approval of the Board of Directors.

Section 5.11 The Treasurer, the Executive Secretary, and all Officers, Agents or Employees authorized by the Board of Directors to endorse or execute drafts for the payment of money, shall give a bond in a penal sum and with sureties approved by the Board of Directors, for the faithful performance of his or their duties, the premiums therefor, to be paid by the Society.

Section 5.12 The Executive Secretary shall be appointed by the Board of Directors under an employment agreement approved by the Board of Directors, fixing his salary, term of employment, and other conditions. The Executive Secretary shall be subject to removal by a vote of two-thirds (2/3) of the Board of Directors present and voting by secret ballot as a meeting.

Section 5.13 The Executive Secretary shall act as secretary of the Board of Directors and of the Executive Committee and as an ex-officio member of all other committees, except the Nominating Committee. He may take part in the deliberations of all these bodies; but shall not have a vote therein. The Executive Secretary shall, under the supervision of the Finance Committee, have charge of the collections and of keeping the books. He shall present, at the meeting of the Society following the close of the fiscal year a summary of membership enrollment and other pertinent records, and shall perform such other duties as may be assigned to him by the Board of Directors, the Executive Committee, or the President.

Section 5.14 After the close of the fiscal year, the accounts of the Society shall be audited by a certified public accountant approved by the Board of Directors, and the auditor's report shall be presented by the Treasurer at the next Meeting of the Society, and shall be published in the official publication.

Section 5.15 Unless waived in writing or by telegraph or by cable, notice of any regular or special meeting of the Board of Directors shall be given in writing, mailed to the last known address of each member of the Board of Directors, by the Executive Secretary or the President, or the three (3) members of the Board of Directors calling the meeting, not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting.

ARTICLE VI—Elections

Section 6.1 Voting at any meeting may be in person or by proxy, but only voting members in good standing of the Society shall be eligible to act as proxies. A proxy shall not be valid after more than three (3) months from its date of execution. Voting for election of officers, Board of Directors members, on proposals to amend these By-laws, and on questions referred to the Society pursuant to Article V *Section* 5.3 shall be by secret ballot. In the event of any tie vote, the Board of Directors shall decide the vote.

Section 6.2 Together with notice of the Annual Meeting, the Executive Secretary shall forward appropriate proxies and ballots to members entitled to vote. The proxies and ballots shall contain spaces for write-in names.

Section 6.3 The polls for election shall be opened at the opening of the Annual Meeting and shall remain open for a period of five (5) hours. Thereafter the ballots shall be opened by three (3) inspectors of election appointed by the President, who shall be authorized to fill any vacancy occurring among such inspectors. The inspectors of election shall consider ballots and votes to be valid provided the intent of the voter is clear. The result of the vote shall be reported by the inspectors of election in writing, and shall be announced by the President on the second day of the Annual Meeting, whereupon the terms of the inspectors of election shall expire.
The elected candidates shall be installed during the Annual Meeting.

Section 6.4 There shall be published in the official publication of the Society, three months prior to the Annual Meeting, the names and qualifications of the consenting nominees as submitted by the Nominating Committee; and two months prior to the Annual Meeting, the names and qualifications of the consenting nominees submitted in proper time by any petitioning group.

Section 6.5 The President, Vice Presidents, and the Treasurer shall each be elected to serve in their respective offices for a term of one (1) year. The Directors shall each be elected for a term of three (3) years, one-third (1/3) of them to be elected each year, or as specified in Article V *Section* 5.1

ARTICLE VII—Meetings

Section 7.1 The Annual Meeting of the Society shall commence on a day and at a time fixed by the Board of Directors, at approximately the middle of the calendar year, and shall continue from day to day until adjourned.

Semi-Annual Meetings shall be held at such times as may be fixed by the Board of Directors. Special Meetings may be called at any time by the Board of Directors, and shall be called by the Board of Directors upon the written request of the President, or fifty (50) members of the Society. Meetings shall be held at such place or places as the Board of Directors may designate, and shall be governed by the laws of the State of New York, the Certificate of Consolidation, and these By-laws. At any meeting of the Society, the presence of fifty (50) members entitled to vote in person or by proxy shall constitute a quorum.

Section 7.2 Notice of the Annual and Semi-Annual Meetings of Society shall be published in the official publication of the Society. Notice of any Special Meeting of the Society shall be given in writing by the Executive Secretary and mailed, postage prepaid, not less than twenty (20) nor more than forty (40) days before the date fixed for the meeting, to each member of the Society at his last known address appearing on the records of the Society.

Notice of a Special Meeting shall state the purpose for which the meeting is called, and no business other than that set forth in the notice shall be entertained or transacted thereat.

ARTICLE VIII—Committees

Section 8.1 Committees of the Society shall consist of General Committees and Special Committees.

Section 8.2 Unless otherwise provided, the General Committees, and the respective Chairmen thereof, shall be designated by the President with the approval of the Board of Directors as soon as practicable after the close of the Annual Meeting of the Society.

Section 8.3 The Board of Directors shall prescribe the qualifications of members of committees and the number of committees. It may in addition, unless otherwise provided, adopt rules specifying the size of committees, the length of term members may serve, when members may be reappointed, selection procedure, and approval of appointments.

Section 8.4 The Board of Directors may from time to time create other committees of one (1) or more members, and define their powers and duties, and it may abolish any such committees.

Section 8.5 The Chairman and Vice Chairman of each General or Special Committee shall be a Member or Associate Member, except as otherwise provided herein.

Section 8.6 The President may appoint any person or persons to serve in a consulting capacity to any General or Special Committee.

Section 8.7 The names of the General Committees of the Society shall be as follows:

Executive	Education
Finance	Honors and Awards
General & Administrative Coordinating	International Relations
Technical Coordinating	Meetings Arrangements
Divisional Advisory	Membership Development
Regions Central	Nominating
Chapters Regional	Professional Development
Long-Range Planning	Program
Advisory Board—Past Presidents	Publications
Admissions and Advancement	Public Relations
Charter & By-laws	Research & Technical
Exposition	Standards
	Guide & Data Book

Section 8.8 The duties and functions of each of the Committees shall be as follows:

Section 8.8.1 The Executive Committee shall consist of the President, who shall be its Chairman, the immediate Past President, the 1st Vice President, the 2nd Vice President, and the Treasurer. It shall meet at the call of the President, or upon request of any two (2) members of the committee.

It shall investigate and make reports and recommendations to the Board of Directors regarding matters relating to the Society or any member or members thereof. During intervals between Board of Directors meetings, the Executive Committee shall exercise administrative powers of the Board of Directors. Matters of policy determined by the Executive Committee between meetings

of the Board of Directors shall be submitted to the Board of Directors at its next meeting for ratification.

Section 8.8.2 The Finance Committee shall consist of five (5) Members including the Treasurer, one Vice President and three (3) others, each of these three (3) to serve a term of three (3) years. The President shall appoint one of these last three (3) committeemen each year, shall select the Vice President who is to serve, and shall fill any vacancies. One of three-year appointees shall be a member of the Board of Directors.

Section 8.8.3 The General and Administrative Coordinating Committee shall consist of the Chairmen and Vice Chairmen of the following committees:

Admissions and Advancement, Education, Honors and Awards, Meetings Arrangements, Membership Development, Professional Development, and Public Relations; and shall have a Vice President as Chairman. It shall be its duty to promote and coordinate participation by the members in the activities of the Society falling within the purview of said committees.

Section 8.8.4 The Technical Coordinating Committee shall consist of the Chairmen and Vice Chairmen of the Program, Publications, Research and Technical, Standards and Guide and Data Book Committees, and shall have a Vice-President as Chairman. In addition, there shall be three (3) Members-at-Large appointed by the President of the Society. It shall be the duty of the Committee to coordinate the activities of the committees named in this Section.

Section 8.8.5 The Divisional Advisory Committee shall consist of the nine (9) Members-at-Large of the Board of Directors, three (3) each being elected from the (1) heating, (2) refrigeration, (3) air conditioning and ventilation areas of membership professional interest, and each of whose major interests shall be in the particular category represented. It shall be the duty of the Committee to review, and advise the Board of Directors, when it deems that any area of membership professional interest is not being duly represented or recognized.

It is the duty of this Committee to report in detail at each meeting of the Board of Directors the degree to which each area of professional interest is being served or recognized in the activities of the Society.

Wherever twenty-five (25) or more members in a Chapter, or two hundred (200) or more members nationally signify their desire to carry out a program at Chapter level, or national levels respectively, in a specialty within the fields defined in *Section* 1.2, this Committee and the Board of Directors shall provide all reasonable facilities customarily provided to other specialties.

Section 8.8.6 The Regions Central Committee shall consist of the 2nd Vice President, as its Chairman, and the Regional Directors. The said committee shall consider and report to the Board of Directors on the activities of Chapters and make recommendations to the Board of Directors concerning the policies, procedures and operation of the Society and its Chapters; it shall coordinate the activities of the Chapters Regional Committees; and it shall investigate applications for the creation of Chapters, and report thereon to the Board of Directors.

Section 8.8.7 The Advisory Board, consisting of the Past Presidents shall advise the Society on matters of basic Society policy or interest.

Section 8.8.8 Chapters Regional Committees, each serving one Regional Area, and each consisting of the Regional Director for the area and one (1) member and one (1) alternate member selected by each Chapter therein, to serve for a term of one (1) year. The said committees shall solicit from the Chapters and Branches within their respective Regional Areas recommendations concerning the policies, procedures, and operation of the Society, its Chapters and Branches, review the same, and make recommendations thereon to the Regions Central Committee. Said committees shall select the Members and alternates to serve on the Nominating Committee, and duly notify the Executive Secretary of such selections. The alternate members of Chapters Regional Committees may be present at committee meetings and participate in the deliberations thereof, but shall not vote therein except in the absence of the committee members for whom they respectively are alternates. The said committees shall hold committee meetings not earlier than forty-five (45) days after the Annual Meeting. Each Regional Director shall be the Chairman of the Committee serving his Regional Area.

Section 8.8.9 The Long Range Planning Committee shall consistently make the necessary studies to prepare for and recommend to the Board of Directors, long range planning on the aims and activities of the Society which in the opinion of the Committee would affect the future welfare and growth of the Society.

Section 8.8.10 The Admissions and Advancement Committee shall receive all applications for membership and advancement and recommend the names of all applicants and the grade for which it considers them to be eligible, except membership grades conferred as an honor, and report to the Board of Directors. The correspondence, information obtained, and proceedings of said committee shall be secret and confidential, and its records concerning unsuccessful applicants shall be destroyed within a reasonable time.

Section 8.8.11 Charter and By-laws Committee, consisting of three (3) Members, which shall consider all matters requiring possible changes in the Certificate of Consolidation, By-laws, Rules and Regulations, and make recommendations thereon to the Board of Directors.

Section 8.8.12 The Education Committee shall advise the Society in its educational programs, in its relations with educational institutions, and those matters pertaining to the requirements for membership of students and graduates therefrom, and related matters.

Section 8.8.13 The Exposition Committee shall coordinate the activities of the Society with expositions, shows, and the like, related to the arts of heating, refrigeration, air conditioning and ventilation.

Section 8.8.14 The Honors and Awards Committee shall recommend to the Board of Directors the candidates for all awards for contributions to the sciences and arts of heating, refrigeration, air conditioning and ventilation, or closely allied fields, and for articles appearing in the official publication, and other gifts or awards including membership grades conferred as an honor. The committee shall include a Past President.

Section 8.8.15 The International Relations Committee shall represent the Society in its cooperation and relationship with foreign engineering societies and groups and make recommendations thereon to the Board of Directors.

Section 8.8.16 The Meetings Arrangements Committee shall study the suitability of locations for meetings of the Society and determine that proper facilities are available, and shall make its recommendations to the Board of Directors.

Section 8.8.17 The Membership Development Committee shall publicize the aims, activities, achievements and scientific and educational purposes of the Society toward the end that persons duly qualified shall apply for membership therein.

Section 8.8.18 The Nominating Committee shall consist of eighteen (18) members, each of whom shall hold the grade of Member, Fellow, or Life Member in the Society and shall have been a full Member in good standing in the Society for a period of at least five (5) years at their selection as follows: One (1) member with one (1) alternate from each region of the Society selected by the Chapters Regional Committees; six (6) members with six (6) alternates selected by the Board of Directors of whom two (2) members shall represent each of the three (3) major areas of membership interest, namely, (1) heating, (2) refrigeration, (3) air conditioning and ventilation; one (1) member selected by the President from the last prior Nominating Committee, and the last preceding Past President who will not be a member of the Board of Directors during the year of service of the committee, and who agrees to serve, shall be its Chairman and shall have the right to vote.

The Nominating Committee shall serve during the Society year, except that the term of service shall begin with the opening of the Annual Meeting. Members and alternates shall be selected during the Society year preceding their year of service, as follows: Regional selectees by the Chapter Regional Committees at their regular called meetings, Board of Director selectees by the Board at their Semi-Annual meeting, and the presidential selectee, by the President of the Society at the time of the Society's Semi-Annual Meeting, with all selections becoming effective on the opening day of the Annual Meeting.

There shall be no more than one (1) member and one (1) alternate from any one (1) Chapter, no more than three (3) members and three (3) alternates from any one (1) Region on the Nominating Committee. No member of the Board of Directors shall be eligible to serve on the Nominating Committee.

Nominations of officers and members of the Board of Directors, other than those nominated by the Nominating Committee, may be made in writing by not less than fifty (50) members eligible to vote, upon presentation of such nominations, with each nominee's consent, to the Executive Secretary at least sixty (60) days prior to the first session of the Annual Meeting, whereupon the nominees' names shall be placed upon the ballot with a notation that they are presented by members independent of the Nominating Committee.

Section 8.8.19 The Professional Development Committee shall promote the professional development of the members amid the profession of engineering in general. Without engaging in propaganda or otherwise influencing or intending to influence legislation, this Committee shall endeavor to obtain greater public recognition of the profession of engineering in the advancement of human welfare.

Section 8.8.20 The Program Committee shall plan the general character of all technical meetings of the Society, and shall solicit, receive, and select papers for presentation at such meetings.

Section 8.8.21 The Publications Committee, subject to the direction of the Board of Directors, shall formulate the editorial policies of the Society and all of its publications. The chairman may appoint sub-committees of one (1) or more members to review and report to the committee on the quality and appropriateness for publication of papers and bulletins intended for

presentation or presented at Society meetings and the discussions thereof. In the performance of its functions the said committee and its sub-committees shall be subject to the following conditions: (A) that the data recommended for publication shall tend toward the professional education of the individual engineer; (B) that such data shall be free from commercial bias; and (C) that such data shall tend to advance for the public benefit the sciences relating to the arts of heating, refrigeration, and air conditioning and ventilation, or the allied arts and sciences. The Chairman of the Guide and Data Book Committee shall be an ex-officio member of the Publications Committee and the Chairman of the Publications Committee shall be an ex-officio member of the Guide and Data Book Committee.

Section 8.8.22 The Public Relations Committee shall conceive and direct a program of public relations to make fully known and easily understood the aims, activities and achievements of the Society as well as its scientific and educational purposes with the object of cultivating and stimulating the interest of members, other professional people and the general public in the Society and its affairs.

Section 8.8.23 The Research and Technical Committee, subject to the direction of the Board of Directors, shall conduct and coordinate basic research and technical studies in the fields of heating, refrigeration, and air conditioning and ventilation, subject to the proviso that these activities shall be devoted to the public welfare and general benefit, and shall not be designed to promote any individual, private, or commercial interests.

In addition to the research activities, this committee shall plan for and have charge of the activities of the technical committees appointed to further the advancement of the arts and sciences of heating, refrigeration, and air conditioning and ventilation, and the allied arts and sciences for the public benefit. It shall determine the scope of the activities of each of these technical committees.

The committee shall consist of twelve (12) members nominated by the Board of Directors, and elected by vote of the Board of Directors. Four (4) members shall be elected each year during the Semi-Annual Meeting of the Society to serve for a term of three (3) years commencing with the next Annual Meeting of the Society.

The Chairman, on the recommendation of the Committee, shall appoint such Technical Committees as may be deemed expedient to carry out the objectives of the Committee, or to advise it on specific projects.

Section 8.8.24 The Standards Committee shall have charge of the selection, development, preparation and submittal to the Board of Directors of all codes and standards in the fields of heating, refrigeration, and air conditioning and ventilating engineering, and all revisions thereof, to be considered for adoption. It shall cooperate with other organizations in the development, preparation, and adoption of codes and standards.

Following approval by the Standards Committee of a code, standard, or a revision thereof, a notice that such is available for comment shall be published in the Society Journal. This notice shall state that preliminary copies will be sent upon request and that comments addressed to the Standards Committee will be received for a period of sixty (60) days following publication of the notice. The Standards Committee, after reviewing such comments, shall make final recommendations to the Board of Directors for adoption of the code, standard, or revision thereof. All written objections which have been overruled by the Standards Committee shall be submitted to the Board of Directors.

Adoption of a code or a standard, or a revision thereof, shall require the approval of the Board of Directors, and the Board of Directors shall assure that proper consideration has been given to it.

Following approval by the Board of Directors, a notice of such approval and availability and effective date of the code, standard, or revision thereof, shall be published in the Society Journal.

The activities of the Committee shall be solely for the development of engineering science, and the Committee shall not engage in influencing enactment of building or other codes, or in propaganda, or other activities designed to influence legislation.

Section 8.8.25 The Guide and Data Book Committee shall consist of a Chairman and two subcommittees, and shall have the responsibility of preparing two volumes of the Guide and Data Book: one volume entitled Fundamentals and Equipment, and one volume entitled Applications, published in alternate years. Each subcommittee shall have responsibility for one volume and shall consist of six members, one of whom shall be designated as Vice-Chairman.

Upon completion of a volume a new subcommittee shall be appointed to prepare the next revision of that volume. At least three (3) members of the new subcommittee shall not have served on the previous subcommittee for that volume.

Section 8.9 All General Committees and Special Committees, except the Nominating Committee shall render to the Board of Directors, prior to the Annual and Semi-Annual Meetings of the Society, annual reports of their activities and shall submit progress reports at other times on request of the President.

Section 8.10 Any Committee may be permitted by the President, with the approval of the Board of Directors, to coordinate its activities with other organizations or groups having interests kindred to those of the Society.

Section 8.11 Each General Committee at the request of the Board of Directors shall prepare an Operational Procedure which shall govern its activities after approval by the Board of Directors. These may be amended as required, with the approval of the Board of Directors.

Section 8.12 The Board of Directors may by a two-thirds (2/3) vote remove a member of any committee excepting Regions Central Committee, Chapters Regional Committee and Nominating Committee.

Section 8.13 Each committee's actions, proceedings, findings, conclusions and reports shall be subject to the direction and review of the Board of Directors, and the Board of Directors may take such steps, or see that such steps are taken by the committees as may be appropriate to comply with the Charter and By-laws, and to make effective any resolution adopted by the Society or any resolution, rule or directive of the Board of Directors.

Section 8.14 If any doubt or controversy should arise as to whether a particular subject or matter is within the jurisdiction of a committee or whether any action should be taken by a committee, or in the case of a committee tie vote, the same shall be settled and determined by the Board of Directors.

ARTICLE IX—Chapters and Regions

Section 9.1 The Board of Directors may establish Chapters which shall operate under the provisions of the Charter and the By-laws of the Society, and the Rules and Regulations of the Board of Directors.

Section 9.2 Branches of the Society may be established, operated, and maintained under the direction and in the discretion of the Board of Directors. Such Branches shall be Special Branches, consisting of groups within continental North America; Overseas Branches, consisting of groups outside of continental North America, and Student Branches.

Section 9.3 Chapters of the Society shall be grouped in geographical areas by the Board of Directors and each such area shall be designated as a Region. The number and delineations of Regions and changes therein shall be published in the official publication.

Section 9.4 A Chapter or Branch of the Society may be authorized upon approval by the Board of Directors of a written petition of twenty-five (25) members of the Society and the adoption of By-laws, based on the Model By-laws for Chapters or Branches, which have been approved by the Board of Directors.

Only members of the Society in good standing shall be eligible to become and remain Chapter or Branch members. Chapter or Branch members shall hold the same grade of membership in the Chapter or Branch as are held by them in the Society. No member shall vote or hold office concurrently in more than one (1) Chapter or Branch of the Society. All grades of Chapter or Branch members, except Students, shall be eligible to vote and hold office in Chapters or Branches.

Section 9.5 A charter shall be granted by the Board of Directors to each duly authorized Chapter or Branch. It shall be signed by the President and attested by the Executive Secretary.

Section 9.6 The elected officers of Chapters or Branches shall receive no salary, emolument, or compensation for their services as such. Chapters or Branches shall not act for the Society or subject the Society to any financial or other obligation, except such as the Society or the Board of Directors may by resolution specifically assume. Notice to the foregoing effects shall be imprinted on the stationery used by each of the Chapters or Branches. Each Chapter or Branch shall promptly file a copy of its Minutes with the Executive Secretary of the Society and make report to said Secretary of all of its proceedings. Each Chapter and Branch shall file with the Chairman of its respective Chapters Regional Committee its recommendations concerning the policies, procedures and operation of the Society, its Chapters and Branches. No contributions, except dues, and assessments, shall be solicited by Chapters or Branches without the written approval of the Board of Directors. Chapters or Branches shall not issue publications, other than Chapter rosters, or use the Society's name or emblem or Chapter or Branch insignia, without the approval of the Board of Directors. Chapters or Branches shall give no recommendations, endorsements or approvals of any scientific, literary, mechanical or engineering product for the promotion of private interests.

Section 9.7 Each Chapter or Branch shall annually elect and install officers prior to the Annual Meeting of the Society. Each Chapter may be provided with funds by the Society as the Board of Directors may deem appropriate.

Section 9.8 The charters of Chapters, Special Branches and Student Branches, and Overseas Branches, may be revoked by a two-thirds (2/3) vote of all the members of the Board of Directors after written preferment of charges, sixty (60) days written notice of hearing sent by

registered mail to the President of the Chapter or Branch, and an adequate opportunity to be heard before the Board of Directors or a committee of three (3) or more Members designated by the Board of Directors.

ARTICLE X—Funds

Section 10.1 Society Reserve Fund. Admission fees and such other funds as may from time to time be recommended by the Finance Committee and allocated by the Board of Directors shall be set aside and the principal thereof maintained as a Society Reserve Fund. The Board of Directors is authorized and empowered, in any fiscal year in which the Society's revenues may be insufficient to meet expenses, to utilize a maximum of twenty percent (20%) of the Society Reserve Fund as valued on the first day of the fiscal year in which such a withdrawal may be required.

Section 10.2 Allocation of Dues for Research. Unless changed by the Society at an Annual or Special Meeting, a percentage determined by the Board of Directors of the dues shall be allocated for basic or fundamental research in the principles and laws underlying matters in the arts relating to the sciences of heating, refrigerating, and air conditioning and ventilation, and the allied arts and sciences.

Section 10.3 Investment of all funds of the Society shall be made by the Treasurer upon direction of the Finance Committee or the Board of Directors, or under such other arrangements as are approved by the Board of Directors.

ARTICLE XI—Amendments

Section 11.1 Prerequisites. These By-laws may be amended by a two-thirds (2/3) vote of the Society at an Annual or Semi-Annual Meeting thereof, provided that written notice of the proposed amendment, subscribed by two-thirds (2/3) of the members of the Board of Directors or by fifty (50) Members, be given at a previous stated or Special Meeting, and that notice thereof as pertinently amended by majority vote at said stated or Special Meeting be also given by the Executive Secretary in the notice of the Annual or Semi-Annual Meeting.

Section 11.2 Renumbering. The Board of Directors may, by a two-thirds (2/3) vote, renumber existing articles or Sections of these By-laws.

ARTICLE XII—Interim Provisions

Inasmuch as this Society is a corporation formed by the consolidation of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and *The American Society of Refrigerating Engineers*, known as the constituent corporations, and as ASHAE and *ASRE* respectively; and
inasmuch as the members of the constituent Societies desire to continue the officers and directors of the respective constituent corporations for the welfare and benefit of the consolidated corporation and its members; and
Notwithstanding any provisions of these By-laws of this consolidated corporation to the contrary, it is understood and agreed that the By-laws of this consolidated corporation shall include the following interim provisions:

(*A*) The officers and directors shall be persons named and for the terms specified in the schedule which is made a part of the agreement for consolidation of the constituent Societies, and which schedule is made a part hereof;

(*B*) Upon the death, or inability or failure to act, of any of said officers or directors, his successor for his unexpired term shall be elected at the next meeting of the Board of Directors and the person so elected shall be from the same constituent Society as his predecessor; and

(*C*) After the expiration of the terms of all officers and directors as provided for in said schedule, then these interim provisions and this Article XII shall become null and void.

ACCOUNTANT'S REPORT

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

In accordance with the authority contained in the minutes of the meeting of the Board of Directors on June 12, 1960, we have examined the Consolidated Statement of Financial Condition of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC., New York, N. Y. at June 30, 1961 and the consolidated results of its operations for the year then ended. Our examination was conducted in accordance with generally accepted auditing standards, and accordingly included such tests of the accounting records and such other auditing procedures as we considered necessary in the circumstances.

In our opinion, the accompanying Consolidated Statements of Financial Condition and of Income and Fund Balance of the AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC. present fairly its financial condition at June 30, 1961 and the results of its operations for the year then ended. These statements were prepared in accordance with generally accepted accounting principles and with the directives of the Board of Directors applied on a basis consistent with that of the preceding year.

FRANK G. TUSA & CO.
Certified Public Accountants
Dated September 25, 1961

CONSOLIDATED STATEMENT OF FINANCIAL CONDITION—JUNE 30, 1961

ASSETS

CURRENT		
Cash on Hand and On Deposit		$195,875.06
Accounts Receivable		
Net of Allowance of $1,205.93 for Doubtful Accounts and Agency Commissions	$168,928.40	
Deposits	2,123.00	
Accrued Interest	1,911.14	172,962.54
Inventories—At Lower of Cost or Market		36,459.40
Prepaid Insurance and Dues		4,821.18
Total Current Assets		410,118.18
INVESTMENTS IN SECURITIES—AT COST		
Market Value $526,611.00		462,099.62
PROPERTY AND EQUIPMENT		
Net of Allowance of $28,024.31 for Depreciation		123,992.91
DEFERRED PUBLICATION COST		1,323.49
		$997,534.20

LIABILITIES AND FUND BALANCE

CURRENT LIABILITIES		
Accounts Payable		$199,217.52
Prepaid Dues and Admissions		
Candidates	$4,537.25	
Elected Members	7,237.13	11,774.38
Federal Excise Tax Payable		46.26
Due to Trustees of AMERICAN SOCIETY OF REFRIGERATING ENGINEERS Pension Trust		12,826.06
Deferred Income		
JOURNAL Subscriptions	311.98	
Guide Copy, Research Material and Equipment Sales	400.22	712.20
Total Liabilities		224,576.42
CONSOLIDATED FUND BALANCE		772,957.78
		$997,534.20

CONSOLIDATED STATEMENT OF INCOME AND FUND BALANCE—JUNE 30, 1961

OPERATING INCOME			
Membership Dues		$404,859.15	
Admission Fees		12,207.50	$ 417,066.65
Publications			
Advertising—Gross	$462,122.11		
Less Agency Commissions and Discounts	73,716.13	388,405.98	
Copy Sales		83,244.98	
Subscriptions and Other		16,020.73	487,671.69
Contractual Projects			24,917.78
General and Earmarked Contributions			38,483.18
Exposition Income			162,692.98
Emblem Sales			2,225.15
Total Operating Income			1,133,057.41
OPERATING EXPENSES			
All Salaries		367,766.78	
Prime Cost of Publications	338,992.53		
Publications, Commissions, Promotion and Other	65,329.61	404,322.14	
Travel—Directors and Committees		30,885.20	
Chapter and Host Chapter Allowances and Supplies		39,533.07	
Annual and Semi-Annual Meetings and Exhibits		19,528.66	
Dues		9,494.50	
Certificates, Emblems, Medals and Awards		4,373.42	
Pension and Group Life Insurance		9,322.02	
Payroll Taxes		8,185.12	
Building Operation		12,344.00	
Cooperative Research Agreements		8,833.34	
Fund Raising		5,539.43	
Rent and Electric		24,460.01	
Printing, Stationery and Office Supplies		10,954.29	
Other Administrative Expenses		83,176.99	
Total Operating Expenses			1,038,718.97
EXCESS OF OPERATING INCOME OVER OPERATING EXPENSES			94,338.44
FINANCIAL INCOME			
Interest Earned on Deposits	1,785.00		
Dividends and Interest—Net of Capital Losses	10,278.03		
Foreign Exchange and Sundry Income	766.39	12,829.42	
OTHER DEDUCTIONS			
Awards	1,214.24		
Excess of Pension and Group Life Insurance Payments over Contributions to Pension Fund	2,376.02	3,590.26	
EXCESS OF FINANCIAL INCOME OVER OTHER DEDUCTIONS			9,239.16
CONSOLIDATED NET EXCESS OF INCOME OVER EXPENSES			103,577.60*
CONSOLIDATED FUND BALANCE—JULY 1, 1960			669,380.18
CONSOLIDATED FUND BALANCE—JUNE 30, 1961			$ 772,957.78

*This item includes:

Exposition income (deferred to fiscal 1961-62)	$ 81,346.49
Admission fees to reserve	12,207.50
Interest and dividends	9,239.16
Total	$ 102,793.15
Excess actual income over actual expense	$ 784.45

ASHRAE ANNUAL REPORT COVERING THE FISCAL YEAR 1960-61

A MESSAGE FROM THE PRESIDENT

The fiscal year 1960-61 has been one of the most significant in the 67-year history of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) and its predecessor societies.

This has been a year of important decisions, changing patterns, growth and progress. We have taken a long and analytical look at our membership, our publications, research program, staff administration, financial structure and professional accomplishments. Although the going has been somewhat hazardous at times, I feel strongly that the Society has emerged stronger and better united than ever before in its activities and objectives.

The office of President has had three different occupants, two of whom effectively administered our affairs during this eventful period . . . Walter A. Grant from July, 1960, to February, 1961, and Robert H. Tull from February to July. Mine is the first full-year term since the merger of our two predecessor societies in 1959. This operation on a semiannual basis has been rather difficult. The short periods of the Officers, Board of Directors, duplication of offices and of personnel have made it troublesome to operate on an efficient scale.

In the past, staff administration also presented problems. The staff was divided at three locations, Cleveland and two in New York. During the fiscal year, plans were consolidated to move all of the staff into the new United Engineering Center and the resultant unified operation of 40 personnel is much more convenient and productive. The only staff member not located at headquarters is the midwest advertising representative. Several staff positions have been added to accommodate more completely all aspects of the Society. These include an Associate Secretary, Assistant Secretary-Public Relations and Fund Raising, and Manager of Research.

Probably the most critical issue of the year was the decision by the Board of Directors to expand the ASHRAE research program by grants-in-aid to colleges and private laboratories and to close our laboratory in Cleveland. In existence since 1919, the laboratory had an impressive record of accomplishment and was regarded with pride and affection by many of our members. However, the high overhead, the scarcity of skilled research personnel and the inflexibility of our program made its closing advisable and inevitable. We are confident that our new program will be successful.

Membership dropped slightly, presenting another major aspect of the Society's activities for reappraisal and evaluation. Accordingly, the Membership Development Committee decided to conduct a campaign to add 2000 new members by the end of the 1961-62 fiscal year. Results thus far have exceeded our expectations. As of November 1, 1961, our membership was quite close to the 18,000 mark, making ASHRAE the sixth largest engineering society in the country. During recent months, four new chapters have been added, boosting our number of chapters in the United States and Canada to 92.

There have been considerable changes in our publications. The number of pages devoted to feature articles in our monthly JOURNAL has been expanded by a minimum of 16 and, in practice, has usually exceeded that number. Whenever possible, technical papers given at annual meetings are reported in full and, in many instances, appear prior to the meetings. For the first time, the two widely respected reference sources published by our predecessor societies have been consolidated into one book, which will appear in two volumes . . . the new GUIDE AND DATA BOOK. The 1961 volume, distributed in September, contains data on Fundamentals and Equipment. The second volume, on Applications, will be released in 1962.

Although our advertising revenue has increased almost $50,000 above that for the previous year, creeping inflation and increased expenditures have made it necessary to raise membership dues by 10%. During 1960-61, the Society has had sizable moving expenses, has purchased new office furniture and supplies, has broadened the scope of its staff retirement and pension program and has instituted a public relations program. Additional expenditures in progress now, or contemplated for the future, are slightly higher rental charges for our new headquarters space, our membership campaign and new membership roster.

Currently, we are considering, subject to the concurrence of the Regions Central Committee, the elimination of chapter allowances as of July 1, 1962. This will permit the chapters to have complete autonomy with relation to chapter members, will reduce the cost of operations at headquarters and will both add to our reserve fund and provide us with more money for research.

We are focusing more attention upon our international activities, increasing our affiliation with foreign engineering societies and cooperating with the American Standards Association in formulating international standards.

The location of our staff in the United Engineering Center has brought us into closer relationship with other engineering societies, especially in the areas of public relations and education. We are proud of our new headquarters and cordially invite all members to visit this impressive building, to meet our staff and to become more familiar with our complex operation.

JOHN EVERETTS, JR.
President

RESEARCH

This year, ASHRAE has budgeted $150,500 for research, the bulk of which will be financed by industrv contributions and by income from the biennial International Heating and Air-Conditioning Exposition, sponsored by our Society. Expenses for Research Administration are met from general Society funds.

As the first step in this program, the Society has reorganized its Research and Technical Committee structure. Sixty-nine Technical Committees are now grouped into 10 sections, each section headed by a member of the Research and Technical Committee. It is the task of each Technical Committee to determine the need for research in its area and to assist the Manager of Research in selecting the laboratory facilities best able to carry out the needed investigation. In addition, there are six special Task Groups: Physiological Research and Human Comfort, Contaminants in Refrigeration Systems, Hydronics, Food Refrigeration and Food Technology, Applied Heat Transfer, and Air Conditioning System Control.

Serving on these Committees and Task Forces are approximately 600 experts, in various fields, who donate their time and skills.

Currently, we have made contractual agreements and are negotiating grants-in-aid at the annual rate of $80,000 to such institutions as Kansas State University, University of Illinois, University of Wisconsin, University of Kentucky, Case Institute of Technology, Northwestern University, University of Florida, Columbia University, University of Arizona and University of California.

Last summer, ASHRAE transferred its controlled environment facility from its Cleveland Laboratory to the Kansas State Environmental Research Center, newly established by the Kansas State University of Agriculture and Applied Science at Manhattan, Kansas. The latter will continue the study of controlled environment, carried on by the Society in recent years. Present ASHRAE grants are being utilized by Kansas State for an interim Physiological Research Program and for investigations in heat transfer.

Also, the Society has transferred its Solar Calorimeter with all equipment to the University of Florida, which will continue ASHRAE's study of air-conditioning loads due to insolation. The Solar Calorimeter will be housed in a 12 x 12-ft. structure at the University's Engineering and Industrial Experiment Station at Gainesville. The co-sponsored project is known as the "ASHRAE-University of Florida Solar Calorimeter."

Other projects, now under way or in the process of negotiation, are Steam and Condensate Flow in Pipes, Air-Conditioning System Control, Air Filters and Dust in Homes, Boiling Refrigerants Outside Rotating Cylinders, Heat and Mass Transfer in Dehumidifying Coils, Ventilation Requirement, and Safety of High-Temperature Water Systems.

MEMBERSHIP

During the past six months, the Associate Secretary has devoted a major portion of his time to a study of the procedures utilized in keeping membership records. Our goal has been the improvement of the service rendered to members by headquarters in keeping records up to date and in providing chapter officers with necessary information and materials as rapidly as possible.

To facilitate this, we have instituted

a new IBM system, have developed entirely new record-keeping procedures and have simplified the application form to be filled-in by prospective members.

On September 1, the Membership Development Committee announced its campaign to add 2000 members within the fiscal year 1961-62. With the assistance of the Public Relations Committee, promotional materials were prepared and made available to all chapters. These materials included a new booklet—ASHRAE, YOUR KEY TO PROFESSIONAL ADVANCEMENT, recruitment posters, campaign instructions and monthly report forms. The theme of this campaign is EVERY MEMBER AN ASHRAE BOOSTER.

Several chapters have already increased their membership by the quota assigned to them. Also, the Society has added four new chapters . . . Akron (Region V), Chicago (Region VI), East Texas (Region VIII) and South Dakota (Region IX). It is anticipated that several more new chapters will swell our ranks before the end of the 1961-62 fiscal year.

A Membership Roster, last published in 1959, will be available to all members in early 1962. The new roster will list the name of each member, his grade, year he joined the Society, his business title and address, his home address, and an asterisk to indicate his preferred mailing address. The roster will also contain a separate listing of members by chapters.

PUBLICATIONS

Publications are a major contribution of the Society to members, industries, libraries, schools, the Government, foreign affiliates and institutions and to the U. S. public in general. In addition to the monthly JOURNAL and the annual GUIDE AND DATA BOOK, ASHRAE published its annual TRANSACTIONS, RESEARCH REPORTS, PREPRINTS AND STANDARDS.

A total of 65 technical papers, most of which were presented at annual technical sessions and symposiums, appeared in the 12 1960-61 JOURNALS. This represented an increase of 25% in the number of technical articles which appeared the previous year.

During 1960-61, the JOURNAL was improved in format and typographical style and, just recently, a senior associate editor was added to the editorial staff.

The new consolidated GUIDE AND DATA BOOK incorporates the Heating, Ventilating and Air-Conditioning GUIDE, published from 1922 to 1960, and the Refrigerating DATA BOOK, 1932-1959. The 1961 volume of the two-volume series features 880 pages of reference material on Fundamentals and Equipment. The second volume, on Applications, has been in the planning stage for more than two years and will be published in 1962.

The 1961 Fundamentals and Equipment volume contains sections on Theory, Materials, Load Calculations, System Components, Unitary Refrigeration Equipment, Air Conditioning Units and Refrigerant Systems: Charging, Cleaning and Lubrication. Features of this publication are a complete index for the 67 chapters, numerous charts, graphs and tables and 406 pages of product specifications.

In scope and depth, this volume is the Society's most complete reference book published thus far, containing engineering information essential to the planning, installation and maintenance of climatic control and refrigeration systems.

The 1960 Volume of TRANSACTIONS is the Society's 66th and contains the unabridged technical papers and discussions presented at the 1960 Semiannual Meeting in Dallas and the Sixty-seventh Annual Meeting in Vancouver. Also included are the programs for these meetings, the officers, boards of directors and committee structures for this year, the annual report of the Research and Technical Committee, the financial statement and obituaries.

STANDARDS

Among ASHRAE activities which have a direct and often immediate effect upon American living conditions are Standards established to assist industry and the general public. Society Standards offer a uniform method of testing equipment for rating purposes, suggest safe practices in designing and installing equipment and provide proper definitions of equipment. The creation of ASHRAE Standards is determined by the need for them. Conformance to these Standards, although completely voluntary, is universal.

The principal Standard published by ASHRAE in 1961 was the Method of Testing for Rating Room Air Conditioners. Long desired by industry, this Standard prescribes a method of testing for rating room conditioners in order to obtain cooling capacities and air flow quantities. Another important Standard, now under proposal, is the Measurement of Sound Power Radiated from Heating, Refrigerating and Air-Conditioning Equipment. It is acknowledged that proper sound control with respect to air-conditioning systems cannot be obtained unless there are adequate means for objective measurement of sound produced by this equipment.

Other Standards developed or revised by the Society during 1960-61 include those for: Testing for Rating Water-cooled Refrigerant Condensers; Testing for Rating Liquid Coolers; Testing for Rating Remote Mechanized-draft Air-cooled Evaporative Condensers; Testing for Rating Unitary Air-conditioning Equipment; Testing and Rating Return-line, Low-Vacuum Heating Pumps; Testing for Rating Self-Contained Mechanically Refrigerated Drinking Water Coolers; and Testing for Rating Unitary Heat Pumps for Air Conditioning.

Notable was the adoption of our Designation of Refrigerants Standard as an American Standard through the procedures of the American Standards Association (ASA).

In October, 1960, the Technical Secretary and two ASHRAE Members were U.S. delegates at the Paris Meeting of the International Organization of Standardization. The areas of interest in which these delegates were concerned primarily were the number designation of refrigerants and the testing of factory-assembled air-conditioning units. Several ASHRAE Standards were discussed at this Meeting and were proposed as the bases for future international standards.

ASHRAE PSYCHROMETRIC CHART

Perhaps the Society's most significant technical contribution during the past year is the development of a new psychrometric chart, the result of 50 years of analysis and study.

This new chart shows the degree of moist or dry air as determined from the properties of temperature, heat content, moisture content and air volume. Basically an enthalpy-humidity ratio chart, it shows such factors as dry bulb, wet bulb, dew point, heat

content, humidity ratio and volume information.

NATIONAL MEETINGS

Cognizant of the limitations of time and money, ASHRAE has geographically rotated its annual meetings in order to give more members the opportunity to participate in national activities and to keep abreast of up-to-date professional developments. Since the summer of 1959, the five annual meetings have been held in Lake Placid, N.Y.; Dallas; Vancouver, B.C.; Chicago and Denver. Meetings scheduled for the near future will be held in St. Louis (Jan.-Feb. 1962) and Miami (June 1962).

In February 1961, 3319 members and guests registered for the Semiannual Meeting in Chicago. Concurrently, more than 21,600 visitors attended the 15th International Heating and Air-Conditioning Exposition, sponsored by ASHRAE. Nearly 500 exhibitors displayed their equipment at this show. The Meeting program featured four technical sessions at which 16 papers were presented, five Symposiums and six Forums. Guest of Honor was the Honorable Leslie Rayner, President of the Institution of Heating and Ventilating Engineers, London, England.

In June, at Denver, total attendance for the 68th Annual Meeting exceeded 600. There were three technical sessions at which 15 papers were given, four Symposiums and seven Forums.

A feature of the forthcoming four-day Semiannual Meeting in St. Louis will be 20 research exhibits covering significant educational and research developments in our industry.

In November, ASHRAE participated in a joint session with the American Society of Mechanical Engineers (ASME) on the general subject of psychrometry and in special observance of the 50th anniversary of the development of the rational psychrometric formulae as published by the late Dr. Willis H. Carrier.

HONORS AND AWARDS

In recognition of distinguished service to the Society and to encourage professional development, ASHRAE confers five major awards. All of these awards were presented in June 1961 at the 68th Annual Meeting.

The highest award—the F. Paul Anderson Medal—was bestowed on J. Donald Kroeker, President of J. D. Kroeker and Associates, consulting engineers in Portland, Ore. Mr. Kroeker was cited for his outstanding contribution to the advancement of the Society, to that of his chosen profession and to the industry with which his activities were allied.

Winner of the ASHRAE Klixon Award for 1960 was George V. Downing, Jr., Merck Sharp & Dohme Research Laboratories, Rahway, N. J., for his paper, "Thermoelectric Materials," which appeared in the November 1960 issue of the Society JOURNAL. This award is sponsored by the Spencer Thermostat Div of Metals and Controls Corporation for the best paper published in the ASHRAE JOURNAL on a subject relating to electric motors or controls. The Klixon Award consists of $150 and a certificate.

The Wolverine-ASHRAE Diamond Key Award was presented to Bruno F. Morabito for his paper, "How Higher Cooling Coil Differentials Affect System Economics," which appeared in the August 1960 issue of ASHRAE JOURNAL. This award is sponsored by the Wolverine Tube Div of Calumet and Hecla for the best paper published in the ASHRAE JOURNAL, March through February. This award is a gold key with a diamond insert.

The ASHRAE Willis H. Carrier Award was conferred on Carl F. Speich, co-author of the paper, "Acoustical Systems Determine Oil Burner Pulsations and their Amplitudes," presented at the Society's Chicago Semiannual Meeting in February, 1961. This award, $250 and a scroll, is sponsored by the Carrier Corporation for the best paper presented at a national meeting by an Associate Member under 30 years of age. Mr. Speich is with the Battelle Memorial Institute, Columbus, Ohio.

Alan B. Wagner, student at Case Institute of Technology, Cleveland, Ohio, was presented the ASHAE-Homer Addams Award for 1960. This $600 award is sponsored by the Addams Memorial Fund to be given to a graduate student working on an ASHRAE research project. The R and T Committee selects the school and the school recommends the student.

In addition to the above awards and to the presentation of Fellow and Life Member Certificates, ASHRAE conferred an Honorary Membership on Col. Crosby Field, Presidential Member of the Society and President of Flakice Corporation.

Also, the Society presented a scroll to Cloud Wampler, Chairman of the Board and Chief Executive Officer, Carrier Corporation, commemorating the 50th anniversary of Dr. Willis H. Carrier's publication of "Rational Psychrometric Formulae." These formulae are the foundation of all current air-conditioning design and the bases for ASHRAE's present Psychrometric Chart.

PUBLIC RELATIONS

The ASHRAE public relations program picked up momentum in 1960 with the staff addition of an Assistant Secretary–Public Relations and Fund Raising. In 1961, a change was effected in the Society By-laws to broaden the scope and responsibilities of the Public Relations Committee. The newly written by-law (Sec. 8.8.22) now reads, "The Public Relations Committee shall conceive and direct a program of public relations to make fully known and easily understood the aims, activities and achievements of the Society as well as its scientific and educational purposes with the object of cultivating and stimulating the interest of members, other professional people and the general public in the Society and its affairs."

Although the chief emphasis was on improving internal communications and public relations, considerable effort was devoted to general and trade press coverage, including holding a well-attended press conference at the Chicago Semiannual Meeting, held February 13-16.

EDUCATION

Decreasing enrollments in technical institutes offering courses in refrigeration and air conditioning indicate a need for leadership by ASHRAE in stimulating interest in these curricula. This problem is now being investigated by the Education Committee which, with the cooperation of the Public Relations Committee, is also studying ways to make university instructors and students more aware of the career potential in our specific engineering fields.

In connection with these investigations, a "Procedure and Guide" for establishing scholarships by local ASHRAE chapters has been prepared. It is hoped that this new instruction form will help to increase the number of chapter scholarships.

In the planning stage is a revision of OPPORTUNITIES IN EN-

GINEERING, which has been used widely by high schools and guidance counselors throughout the country in acquainting students with career opportunities in our field and the college or technical training required.

During April and May, 1962, Dr. Rudolph Plank, world-renowned German authority on refrigeration, will visit the United States as a guest of ASHRAE, in conjunction with the Visiting Engineers Program sponsored by the National Science Foundation and administered by the Engineers Joint Council. Dr. Plank is scheduled to lecture at Purdue University, the University of Illinois and the University of Minnesota.

ASHRAE MEMBERSHIP JULY 1, 1960 THROUGH JULY 1, 1961

July 1, 1960*		July 1, 1961	
Honorary Members	6	Honorary Members	6
Presidential Members	49	Presidential Members	48
Life Members and Exempt**	440	Fellows	44
Fellows	27	Life Members	367
Members	6,866	Members	6,925
Associate Members	6,951	Associate Members	7,061
Affiliates	2,904	Affiliates	2,983
Students	187	Students	171
	17,430		17,605
		NET GAIN	175

July 1960–July 1961	
Elections and Reinstatements	1,689
Cancellations, Resignations and Deceased	1,514
NET GAIN	175

* Revised figure
** Not separated in 1961 figures

TRANSACTIONS EDITORIAL STAFF

Editor .... Edward R. Searles

Associate Editor .... Eugenia A. Bajster

1733

TRANSACTIONS

of

American Society of Heating Refrigerating and Air-Conditioning Engineers

No. 1733

ASHRAE Semiannual Meeting, 1961

CHICAGO, ILL.

The program for the Semiannual Meeting, held February 13-16, at the Conrad Hilton Hotel, consisted of four Technical Sessions, a Frozen Food Handling Symposium, a Medium Temperature Water Heating Symposium, an Air Conditioning Symposium, an ASHRAE Research and Technical Plans Symposium, a Domestic Refrigerator Engineering Symposium and a Ventilation Symposium. Additionally, there were six Forums. Paralleling this Meeting was the 15th International Heating and Air Conditioning Exposition, sponsored by ASHRAE, which was held at the International Amphitheater.

A total of 16 formally prepared papers was presented at the four Technical Sessions. In accordance with established practice, the full texts of these papers, together with discussions upon them, appear in this volume of TRANSACTIONS.

Registration for this Meeting was 3300 members and guests, close to record-breaking. It was indicated that 21,627 persons attended the Exposition, which was under the direction of the International Exposition Company, New York, N. Y., with E. K. Stevens, manager.

The General Assembly, convened by President Grant at 9:00 a.m. on Monday, February 13, marked the official opening of the Meeting. Following Mr. Grant's opening remarks was a welcoming address by L. K. Warrick, Director of Region VI.

Proposed amendments to the By-laws were announced by Executive Secretary R. C. Cross. President Grant noted that polls were open for the election of officers and vote on By-laws amendments (those announced at the Vancouver Meeting).

The Welcome Luncheon was held at 12:30 p.m. on Monday with J. C. Scott, Vice President of the Illinois Chapter, as Master of Ceremonies. Following the National Anthem and invocation by Rev. Robert W. Tull, Mr. Warrick and General Chairman Harry Gragg extended greetings and welcome. Mr. Scott introduced the Head Table, Past Presidents, notable guests and Chapter officers.

R. A. Line, Chairman of the Program Committee, was introduced and he called upon speakers at Technical Sessions and Symposiums and Forum Moderators to rise and be acknowledged. Leslie Rayner, President of the Institution of Heating and Ventilating Engineers, London, England addressed the Luncheon briefly. Reporting upon the status of the Society, President Grant noted the progress that was made during the past seven months. Mr. Grant announced that election polls would close at 2:00 p.m.

The Banquet took place on Tuesday evening, beginning at 7:30. The invocation was delivered by Rev. Tull. Master of Ceremonies J. S. Kearney introduced the Head Table. President Grant expressed his appreciation to the Illinois Chapter and Region VI for their hospitality. He requested that all Past Presidents and their wives stand and be acknowledged. Also introduced were Mr. and Mrs. Rayner.

Junior Past President D. D. Wile installed the following officers: John E. Dube, Treasurer (succeeding himself); John H. Fox, Second Vice President; John Everetts, Jr., First Vice President; and Robert H. Tull, President.

As his first official duty, President Tull presented a Past President's Pin and framed certificate to Mr. Grant. He also presented him with a gift. Mr. Tull then outlined his program for the Society during his term.

Burgess H. Jennings was the recipient of the Award of Merit of the Life Members Club of ASHRAE, presented by E. K. Campbell. Professor Andrew Hacker of Cornell University, sociologist and political economist, was featured speaker of the evening. A dance followed the Banquet.

Numerous committee meetings were scheduled and held during the course of the gathering. Rounding out the Semiannual Meeting were various social events.

The First Technical Session was convened at 9:30 a.m. on Monday with D. L. Lindsay as Chairman. Four papers were presented at this session.

Held at the same time, the Second Technical Session had C. W. Pollock as Chairman. As scheduled, there were four papers presented here.

Also concurrent was the Frozen Food Handling Symposium where E. J. Robertson was Chairman. There were five speakers at this session. Two papers presented at this Symposium were published in the ASHRAE JOURNAL (Guadagni, April; Sherby, June). In accordance with established practice, Symposium papers are not published in TRANSACTIONS. Presently, no Bulletins are planned for any Symposiums from this Meeting.

R. S. Buchanan was Chairman of the Forums. These were held, two each, on Monday, Tuesday and Wednesday afternoons.

Professor W. E. Fontaine served as Chairman of the Third Technical Session, held at 9:30 a.m. on Tuesday. Before proceeding with the scheduled program, Mr. Cross announced the results of the election. The Report of the Inspectors of Election appears herewith:

REPORT OF INSPECTORS OF ELECTION

Total Valid Votes: 4370

		Votes received
Second Vice President–John H. Fox		4363
Treasurer–John E. Dube		4366
By-laws Amendments		
Proposal	*For*	*Against*
1	4331	39
2	4334	36
3	4289	81
4	4346	24
5	4337	33
6	4284	86
7	4289	81
8	4270	100
9	4327	43
10	4276	94
11	4336	34
12	4346	24

Sherman Loud
William J. Olvany
Andrew T. Boggs, III
Inspectors

Four papers were presented at this session.

Concurrent with this Session was the Medium Temperature Water Heating Symposium, with Lincoln Bouillon as Chairman. Four papers were given at this session; all papers have been published in the JOURNAL (Bird, April; Miller, May; Carlson, June; Harmon, July).

The Air Conditioning Symposium was also held at the same time. W. R. Moll was Chairman and the topic was Humidity, Humidity Control, Air Motion and Movement. There were four papers at this meeting, two of which were published in the JOURNAL (Nevins, July; Signorelli, July).

E. P. Palmatier, Chairman, called the ASHRAE Research and Technical Program Plans Symposium to order on Wednesday morning at 9:30. There were five speakers who presented their viewpoints on the subject.

The Domestic Refrigerator Engineering Conference was held concurrently. E. J. Von Arb was Chairman at this session where six speakers participated. A condensed version of this Symposium appears in the June issue of the JOURNAL.

The Ventilation Symposium, held at 9:30 a.m. on Thursday, J. B. Graham, Chairman, considered the topic Performance Determination in Installed Systems. This Symposium comprised three papers; one paper (Bricker) has been published in the May issue of the JOURNAL.

R. G. Raney was Chairman of the Fourth Technical Session, held concurrently. Four speakers presented papers.

The Semiannual Meeting was adjourned by President Tull at 12:10 p.m. at the final Business Session.

PROGRAM–SEMIANNUAL MEETING

Conrad Hilton Hotel, Chicago, Ill.

February 13-16, 1961

Saturday, February 11

8:00 a.m.	Advertising Committee breakfast-meeting.
9:30 a.m.	Finance Committee.
12:30 p.m.	Executive Committee luncheon-meeting.
2:00 p.m.	Standards Committee.
4:00 p.m.	Regions Central Committee dinner-meeting.

Sunday, February 12

9:00 a.m.	Program Committee.
9:15 a.m.	Exposition Committee.
10:00 a.m.	Research and Technical Committee.
	Advance Registration.
12:00 noon	Board of Directors luncheon-dinner meeting.
2:00 p.m.	Nominating Committee.
3:00 p.m.	Public Relations Committee.
3:00 to 6:00 p.m.	"Hearty" Welcome, Informal Reception.

Monday, February 13

7:30 a.m.	Speakers' Breakfasts: First Technical Session. Second Technical Session. Frozen Food Symposium. Forum Moderators.
8:30 a.m.	REGISTRATION.
9:00 a.m.	GENERAL ASSEMBLY. Opening of the Semiannual Meeting of the Society. Remarks by President Walter A. Grant. Address of Welcome by L. K. Warrick, Director, Region VI. Announcement of Proposed By-laws Amendments by Executive Secretary R. C. Cross. New Business. Polls Open for Election of Officers and Vote on By-laws Amendments.
9:30 a.m.	FIRST TECHNICAL SESSION. Chairman, D. L. Lindsay, Consulting Engr., Wiggs, Walford, Frost & Lindsay. Corrosion Inhibition on Steel Tubes in Low-Pressure Steam Boilers—W. A. Keilbaugh, Head, and F. J. Pocock, Senior Chemist, Chemical Section, Babcock & Wilcox Co., Research Center.
10:10 a.m.	Design of Wet Cell Air Humidifiers—Dr. M. W. First, Consulting Engr.
10:50 a.m.	Metastable State of Water in Relation to Heat Exchangers—B. H. Jennings, Prof. of Mech. Engrg., Northwestern University.
11:30 a.m.	Flow and Heat Exchange Characteristics of Finned Tube Exchangers—Benjamin Gebhart, Assoc. Prof. of Mech. Engrg., Cornell University.
9:30 a.m.	SECOND TECHNICAL SESSION. INSULATION. Chairman, C. W. Pollock, Mgr. Air Cond. & Refrign. Div., Crane Co. Field Laboratory for Heating Studies—D. B. Anderson, Tech. Asst. to V. P.—Sales, G. A. Erickson, Dir. of Tech. Sales Service, and R. R. Leonard, Res. Engr., Wood Conversion Co. and Dr. R. C. Jordan, Head, Mech. Engrg. Dept., Univ. of Minnesota.
10:10 a.m.	Thermal Conductivity of Porous Materials—Dean Melvin Mark, Lowell Tech. Inst. and M. E. Stephenson, Jr., Consulting Engr.
10:50 a.m.	Heat Transfer through Mineral Wool Insulation in Combination with Reflective Surfaces—C. E. Lund, Professor and R. M. Lander, Res. Assoc., Dept. of Mech. Engrg., Univ. of Minnesota.

11:30 a.m. Thermal Effects of Floor Construction—R. D. Cramer, Asst. Prof. of Housing, Dept. of Home Economics and L. W. Neubauer, Assoc. Prof., Dept. of Agric. Engrg., Univ. of California.

9:30 a.m. Frozen Food Handling Symposium.
Chairman, E. J. Robertson, Res. & Tech. Div., Wilson & Co., Inc.
Early Scientific Background—Clifford Evers, Mgr., Commercial Development, International Minerals & Chemical Co.
Effects of Time and Temperature on Frozen Food Quality—D. G. Guadagni, Western Regional Research Lab. (DA).
Handling Frozen Foods at the State Level—Lowell Oranger, Supt., Div. of Foods, Dairies and Standards, Illinois Dept. of Agriculture.
The Industry Position—Edward Sherby, Publisher, Frosted Food Field.
What the Consumer Wants—Dolores Palmer, Home Economist and Special Representative, Frozen Potato Products Institute.

12:00 noon Opening Ceremonies—15th International Heating and Air-Conditioning Exposition.
Including Heating, Refrigeration, Air-Conditioning & Ventilation, International Amphitheatre.*

12:30 p.m. Welcome Luncheon.
Honored Guest: Leslie Rayner, President, The Institution of Heating and Ventilating Engineers, London, Eng.
Master of Ceremonies: J. C. Scott, Illinois Chapter Vice-President.
Presentations and Announcements.

2:00 p.m. Polls Close for Election of Officers and Vote on By-laws Amendments.

2:30 p.m. Forums
Chairman, R. S. Buchanan, Asst. Dir. of Appliance Engrg. and Res., American Motors Corp.
1. Transport Refrigeration.
Moderator: P. R. Achenbach, Chief, Mech. Systems Section, National Bureau of Standards.
2. A New Psychrometric Chart.
Moderator: John Everetts, Jr., ASHRAE 2nd Vice-President, C. S. Leopold, Inc.

2:30 p.m. Honors and Awards Committee.
International Relations Committee.
Meetings Arrangements Committee.
Membership Development Committee.
Publications Committee.
RAC 2 on Mass Transfer.
Standards Committee.
RP on Physiological Research and Human Comfort.
TC—1.9, Refrigerants and Lubricants.

3:30 p.m. RAC 4 on Environment.

4:00 p.m. RP on Odors.

8:30 p.m. Valentine Get-Together Party.

Tuesday, February 14

7:30 a.m. Speakers' Breakfasts:
Third Technical Session.

* *Exposition, International Amphitheatre, 12:00 noon to 10:00 p.m.*

MTW Symposium.
Air Conditioning Symposium.
Forum Moderators.
TC-2.4, Coils and Condensers, breakfast-meeting.

8:30 a.m. REGISTRATION.

9:30 a.m. THIRD TECHNICAL SESSION.
Announcement of Election Results.
REFRIGERATION.
Chairman, W. E. Fontaine, Professor, School of Mech. Engrg., Dir. for Engrg. Research, Ray W. Herrick Labs., Purdue Univ.
Evaluation of Wet Bulb Data for Cooling Equipment Design—L. W. Crow, Consulting Meteorologist.

10:10 a.m. Reaction of Dichlorodifluoromethane with Petroleum Oils—Dr. H. O. Spauschus, Res. Assoc., and G. C. Doderer, Chemist, Major Appliance Div., Gen. Elec. Co.

10:50 a.m. Sizing of Refrigeration System Pipelines for Optimum Economy—D. J. Renwick, Assoc. Prof. of Mech. Engrg., Michigan State University.

11:30 a.m. Graphical Analysis of a Cross Flow Cooling Tower—Prof. Hideo Uchida, Dept. of Mech. Engrg., Univ. of Tokyo.

9:30 a.m. MEDIUM TEMPERATURE WATER HEATING SYMPOSIUM.
Chairman, Lincoln Bouillon, Bouillon, Griffith, Christofferson & Schairer.
Pipe and Pump Size Reduction with MTW Systems—Homer Bird, Bird, Bird, and Associates.
Heat Transfer—Effect of Increased Temperature Drop on Valve Control—S. W. Miller, Chief Engr., J. J. Nesbitt, Inc.
Effect of System Temperature on Pump Curve and Pressure-Drop Curve—G. F. Carlson, Chief Engr., Specialty Div., Bell & Gossett, Co.
Pressurization of MTW Systems—Raymond Harmon, Consulting Engr., Harmon & Beckett.

9:30 a.m. AIR CONDITIONING SYMPOSIUM.
AIR MOVEMENT, HUMIDITY, AND HUMIDITY CONTROL.
Chairman, W. R. Moll, Sales Mgr., Air-Cond. & Dehumidifier Sales-to-Sears, Whirlpool Corp.
Comfort and Physiological Adjustment of People to Environment—Prof. M. K. Fahnestock, Dept. of Mech. Engrg., Univ. of Illinois.
Air Motion and Movement—Dr. R. G. Nevins, Dept. of Mech. Engrg., Kansas State Univ.
Humidity and its Control in Residential Air-Conditioning Systems—Richard Signorelli, Asst. to V.P. of Manufacture, Mueller Climatrol.
Control of Humidity on Window-type Air Conditioners—F. T. Appel, Projects Mgr., Air-Cond. Div., Whirlpool Corp.

12:30 p.m. ASHRAE Life Members Club Luncheon.

1:00 p.m. Panel on Food Refrigeration and Technology.

2:00 p.m. FORUMS.
Chairman R. S. Buchanan.
3. Better Environments for Learning.
Moderator: J. D. Kroeker, Consulting Engr., J. D. Kroeker & Associates.
4. Corrosion of Air Conditioner Components.
Moderator: C. O. Hutchinson, Mgr., Market Dev., Industrial Paint Div., The Glidden Co.

2:00 p.m. Education Committee.
Research Fund Raising Committee.

	Fuel Consumption.
	Research Exhibit.
	TC-2.8, Air Cleaning Equipment.
	TC-4.6, Heat Pumps.
	TC-1.5, Meteorology and Weather Data.
2:45 p.m.	RP on Hydronics.
6:30 p.m.	Cocktail Party.
7:30 p.m.	Banquet.
	Installation of Officers.
	Awards and Presentations.
	Entertainment and Dancing.

Wednesday, February 15

7:30 a.m.	Speakers' Breakfasts:
	ASHRAE Res. & Tech. Symposium.
	Domestic Refrig. Engrg. Symposium.
	Forum Moderators.
8:30 a.m.	Registration.
9:30 a.m.	ASHRAE Research and Technical Program Plans Symposium.
	Chairman, E. P. Palmatier Mgr., Trans. Equip. Dept., Carrier Corp.
	ASHRAE Research Laboratory Program.
	Environment—B. H. Jennings, Dir.
	Odors—W. F. Kerka, Res. Engr.
	Sound—W. F. Kerka.
	Solar Radiation on Fenestration— L. F. Schutrum, Res. Supervisor.
	Entrance Infiltration—L. F. Schutrum.
	Human and Animal Calorimeter—R. G. Huebscher, Res. Supervisor.
	Control—R. G. Huebscher.
	Flow of Steam in Pipes—C. M. Humphreys, Asst. Director.
10:30 a.m.	How a New Product Comes into Being—W. F. Wischmeyer, Sporlan Valve Co.
	Cooperative Research Programs.
	Review—E. P. Palmatier.
	Benefits—Dr. R. G. Nevins, Dept. of Mech. Engrg., Kansas State Univ.
11:00 a.m.	Benefits of ASHRAE Research to Industry and the Engrg. Profession.
	G. R. Munger, Owens-Corning Fiberglas.
	R. M. Stern, Consulting Engr., Stern & Towne.
	F. H. Faust, General Electric Co.
11:25 a.m.	Filling Out the ASHRAE Laboratory Picture—E. P. Palmatier.
	Physical Plant; Personnel; Finances; Administration, including function of R & T Committee.
11:30 a.m.	Question and Answer Period.
9:30 a.m.	Domestic Refrigerator Engrg. Symposium.
	Progress of Thin Wall in Cabinet Construction, Insulation.
	Chairman, E. J. Von Arb, V.P. in Chg. of Engrg., Revco, Inc.
	Practical Application of Foamed Insulation in Industry—C. W. Newhall, Martin Sweets Co.; F. Hermanns, Admiral Equipment Co.; and W. J. Neary, Jennings Machine Co.
	Bagged Insulation—Robert Bilek, Hotpoint Co.

Theoretical Aspects of Foam Insulation—P. A. Sanguinetti, Mobay Chem. Co. (rep. S.P.I.).
Thin Wall Construction in Refrigerators—Severn Joyce, Marion Hollingsworth and E. B. Frankenhoff, Owens-Corning Fiberglas Co.

10:00 a.m. TC-6.2, Acoustics and Vibration Control.
TC-6.4, Industrial Environment.

1:00 p.m. Publications.
TC-1.6, Combustion and Fuels.
Tour of Dresden Nuclear Power Station, Morris.

2:00 p.m. FORUMS
Chairman R. S. Buchanan.
5. Aluminum Brazing and Welding Techniques.
Moderator: A. J. Haygood, Alcoa.
6. Status of Thermoelectric Refrigeration.
Moderator: G. D. Hudelson, Dev. Dept., Carrier Research & Dev. Co.

6:30 p.m. Past-Presidents' dinner.
D. D. Wile, Chairman.

Thursday, February 16

7:30 a.m. Speakers' Breakfasts:
Fourth Technical Session.
Ventilation Symposium.

8:30 a.m. General Administrative and Coordinating Committee.
REGISTRATION.

9:00 a.m. 36-P, Equipment Sound Testing.

9:30 a.m. VENTILATION SYMPOSIUM.
PERFORMANCE DETERMINATION IN INSTALLED SYSTEMS.
Chairman, J. B. Graham, Dir. of Research, Buffalo Forge Co.
Field Check of Ventilation and Air Conditioning Systems—James Bricker, C. S. Leopold, Inc.
Industrial Ventilation Systems—K. E. Robinson, Personal Hygiene Dept., General Motors Corp.
Ventilation Systems with Special Requirements—R. J. Walker, Plant and Process Engrg. Group Leader, Dow Chemical Co.

9:30 a.m. FOURTH TECHNICAL SESSION.
Chairman, R. G. Raney, Marketing Mgr., Ranco, Inc.
Combustion-Driven Pulsations in Oil-Fired Residential Heating Equipment—A. A. Putnam, Staff Mech. Engr. and C. F. Speich, Principal Mech. Engr., Battelle Memorial Inst.

10:10 a.m. Noise Suppression in Oil Burners—R. W. Sage, Section Head, and H. F. Schroeder, Res. Engr., Prods. Dev. Div., Esso Research & Engrg. Co.

10:50 a.m. Suppression of Pulsations in Oil-Fired Residential Heating Equipment —C. F. Speich, Principal Mech. Engr., and A. A. Putnam, Staff Mech. Engr., Battelle Memorial Inst.

11:30 a.m. Influence of the House on Chimney Draft—A. G. Wilson, Head, Bldg. Services Section, Div. of Bldg. Research, National Research Council of Canada.
ADJOURNMENT OF SEMIANNUAL MEETING OF THE SOCIETY.

12:30 p.m. Board of Directors luncheon-meeting.

HOST CHAPTER
ILLINOIS
OFFICERS

W. V. Richards, *President*
J. C. Scott, *Vice President*
L. H. Streb, *Secretary*
E. R. Teske, *Treasurer*
E. N. Johnson, *Board of Governors*
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E. R. Teske, *Finance*

1734

No. 1734

Corrosion Inhibition on Steel Tubes in Low-Pressure Steam Boilers

W. A. KEILBAUGH F. J. POCOCK

A research program, spanning five years, and under the direction of the Engineering Committee of the Steel Boiler Institute, was undertaken to investigate causes and prevention of tube corrosion in low-pressure steel (15 psig max steam) boilers.

Some of the results from this study were reported in an earlier paper.[1] That report described the testing technique and apparatus in detail and reviewed the work performed from test inception until April 1957.

Discussions with the Engineering Committee of the SBI before the investigation indicated that the principal cause of corrosion was dissolved oxygen in the feedwater and boiler water. Since deaeration of the feedwater is neither practical nor economical for the majority of small boilers, nor is the use of chemical oxygen scavengers feasible, due to control difficulties, the general approach to the corrosion problem was a practical one involving the use of anodic chemical inhibitors.

These earlier tests indicated that two chemical inhibitors merited closer examination. They were the Steel Boiler Institute chromate compound and one other proprietary compound which was a sodium borate-nitrate-nitrite mixture. Additional inhibitors were also investigated.

Consideration was given to the use of high strength low-alloy and copper-bearing steels during both parts of the program. Other investigations[2] had indicated that small amounts of copper, as a steel alloying constituent, would have beneficial effects. Researchers compared the corrosion behavior of these low-alloy steels and plain carbon steel tubes made to

W. A. Keilbaugh is Head, and F. J. Pocock is Senior Chemist, Chemical Section, Babcock and Wilcox Co., Research Center. This paper was prepared for presentation at the ASHRAE Semiannual Meeting, Chicago, Ill., February 13-16, 1961.

ASTM-A 178 Grade A specifications.

Test work was carried out in small boilers operating at essentially atmospheric pressures, in order to duplicate as far as possible, under closely controlled conditions, the field operation of low-pressure steel boilers. Experience has shown that the use of coupon material in laboratory bench-scale tests or in autoclaves does not produce conditions comparable to field operation, since the effect of temperature resulting from heat transfer through the metal cannot be evaluated.

Along with the boiler testing program, the factors influencing corrosion during boiler storage periods were also considered. It is known that many of the corrosion problems of boilers have their inception during storage.

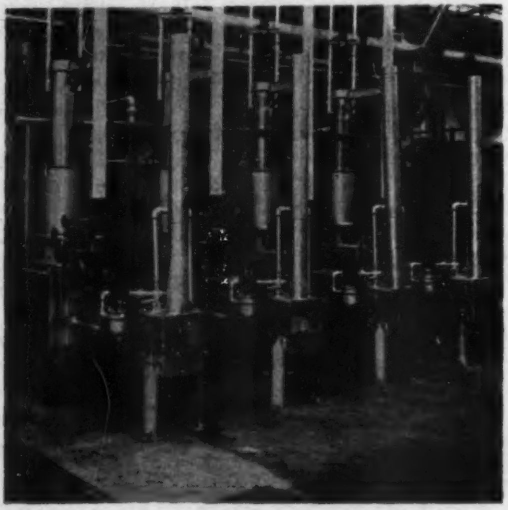
As a result of these tests, a number of recommendations were made regarding the water side care of low-pressure steel boilers. If the recommendations are carefully adhered to, there should be a general increase in operational reliability of these boilers.

ACCELERATED CORROSION TESTS

Test boiler design—The test equipment for this investigation consisted of four gas-fired steel boilers, each of which had its own feedwater and condensate system.

Each test boiler contained, except as noted later, four three-in. OD by 48-in. electric resistance welded boiler tubes made to

Fig. 1 General view of test installation



ASTM-A-178 Grade A specifications. These tubes were rolled into ⅜-in. carbon steel tube sheets, and each boiler was fired with a conventional 120,000 Btu per hr gas burner.

The boilers operated at essentially atmospheric boiling conditions. The steam is condensed and the condensate is passed countercurrently to air flow through a baffled aerating tower. The air-saturated feedwater is then fed automatically to the boiler as required by the boiler water level controls. Total steam flow for each unit is approximately 50 lb/hr. Steam losses are made up automatically from aluminum storage tanks containing distilled water. Using distilled water as a starting point, water chemical characteristics could be varied as desired.

Automatic timing controls were provided so that the boilers operated in pairs for 30-min firing intervals with an equivalent off period. This simulated as closely as possible the periodic operation of the usual heating boiler.

The boilers were operated 24 hours a day in this manner for the duration of each test.

Fig. 2 shows a schematic draw-

Fig. 2 Schematic diagram of the test unit

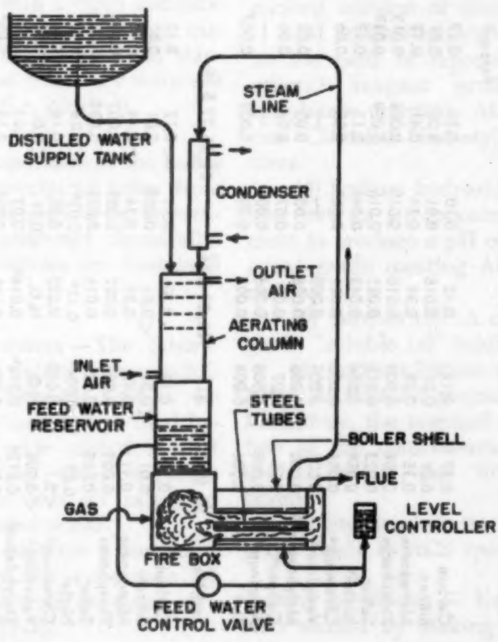


TABLE I—TUBE METAL ANALYSES

Test No.	1 & 5	2	3	4	6	7, 11 & 15	8	9 & 10	12	13[b]	14 & 16	17[b]
						Percent						
Carbon	0.17	0.12	0.12	0.10	0.11	0.17	0.07	0.08	0.12	0.08	0.15	0.11
Silicon	0.01	0.01	0.015	0.01	0.01	0.01	0.01	0.01	0.01	0.36	0.01	<0.01
Manganese	0.35	0.35	0.34	0.34	0.41	0.38	0.38	0.34	0.43	0.29	0.44	0.65
Phosphorus	0.008	0.008	0.010	0.010	0.008	0.013	0.007	0.010	0.010	0.089	0.01	0.010
Sulfur	0.023	0.019	0.021	0.016	0.029	0.037	0.017	0.04	0.05	0.039	0.029	0.027
Nickel[a]	0.059	0.054	0.043	0.045	0.13	0.13	0.13	0.058	0.050	0.28	0.052	1.05
Chromium[a]	0.056	0.052	0.051	0.050	0.05	0.06	0.05	N.D.[c]	N.D.[c]	0.69	0.05	<0.1
Vanadium[a]	0.005	0.005	0.005	—	—	—	—	—	—	—	—	<0.005
Molybdenum[a]	0.008	0.008	0.008	0.008	0.03	0.02	0.03	0.01	0.005	0.008	0.009	0.14
Copper[a]	0.06	0.07	0.08	0.07	0.06	0.06	0.06	0.046	0.064	0.30	0.07	0.98
Aluminum[a]	0.005	0.005	0.005	0.005	—	—	—	0.001	0.002	0.005	0.004	<0.005
Tin[a]	0.006	0.008	0.007	0.007	0.009	0.007	0.008	—	—	—	—	—

Test No.	18, 19 & 20	21[d]	22, 23 & 24	25, 26, 27 & 28			29	30	31	32
Carbon	0.12	0.11	0.10	0.11	0.12	0.11	0.13	0.14	0.16	0.12
Silicon	0.01	<0.01	<0.01	0.008	0.01	0.01	<0.01	<0.01	<0.01	<0.01
Manganese	0.41	0.42	0.39	0.34	0.44	0.40	0.42	0.43	0.44	0.42
Phosphorus	0.009	0.008	0.007	0.008	0.007	0.008	0.006	0.008	0.006	0.007
Sulfur	0.025	0.021	0.016	0.025	0.034	0.023	0.025	0.028	0.027	0.019
Nickel	0.09	0.094[a]	0.13[a]	<0.03[a]	<0.03[a]	<0.03[a]	0.04[a]	0.04[a]	0.04[a]	0.04[a]
Chromium[a]	<0.1	0.010	0.022	0.05	0.16	0.06	0.07	0.07	0.07	0.07
Vanadium[a]	<0.006	N.D.	N.D.	<0.01	<0.01	<0.01	<0.005	<0.005	<0.005	<0.005
Molybdenum	0.012[a]	0.018[a]	0.014[a]	0.02	0.03	0.01	0.03[a]	0.03[a]	0.03[a]	0.03[a]
Copper	0.06[a]	0.37[a]	0.11[a]	0.05[a]	0.17	0.06[a]	0.06[a]	0.06[a]	0.05[a]	0.07[a]
Aluminum[a]	0.005	—	—	<0.01	<0.01	<0.01	<0.004	<0.004	<0.004	<0.004
Tin[a]	—	—	—	<0.01	0.01	<0.01	0.02	0.01	0.02	0.02

[a] Spectrographic Trace Metal Analyses [b] High Strength, Low Alloy Steel [c] N.D. = Not Detected [d] Copper Bearing Steel

ing of a typical test boiler. Fig. 1 shows a general view of the test installation.

Boiler tubes – All those used in these investigations were electric resistance welded. Except for three tests, the tubes were made to ASTM-A-178 Grade A specifications. Three material comparison tests were made, two using high strength, low-alloy tubes and one using copper-bearing steel tubes. One of the three latter tests was reported in the previous paper.[1]

Tubes are received with a thin film of soluble oil as corrosion protection in transit. They are rolled into the tube sheets of the test boilers in this condition, with only light wiping with a cloth and mineral spirits to remove foreign material which is picked up in shipment. In these tests they were not acid cleaned (i.e. pickled).

Chemical composition of the boiler tubes – Representative tubes from each steel heat used in this investigation were analyzed chemically. These steel analyses are contained in Table I.

Base boiler waters – The laboratory distilled water, with a nominal 100 ppm chloride such as sodium chloride added, was used because of the wide variations that occur in natural water supplies. The feedwater was air saturated (at 100 F) distilled water.

Table II contains a summary (tests 1-16) of the conditions of these tests as well as those described here (17-32).

Further tests of the selected chemical inhibitors, in the presence of hardness ($Ca(HCO_3)_2$, $MgSO_4$, $CaSO_4$) in the boiler water, were necessary, since it was felt that these hardness constituents might have some deleterious effect on inhibitor efficiency. Constituents are in most water supplies.

To date, 6 inhibitors have been used in the various test periods. They are as follows:

(1) A buffered chromate compound. Marketed under the auspices of the SBI.

(2) Borate-nitrate-nitrite compound. A proprietary chemical compound distributed by a water treatment company for the express purpose of corrosion inhibition in cooling systems of diesel engines.

(3) Sodium molybdate. Tested on the basis of reported work of others,[2] reagent grade sodium molybdate meeting ACS (American Chemical Society) specifications.

(4) Sodium hydroxide. Used in two tests in a concentration sufficient to produce a pH of 11.0. Reagent grade meeting ACS specifications.

(5) Soluble oil. A commercial grade "soluble oil" inhibitor.

(6) Sodium borate-sodium nitrite. A mixture prepared at the laboratory, the nominal concentration of the constituents was 90% sodium nitrite and 10% sodium borate.

(7) Sodium nitrite. Reagent grade made to ACS specifications.

Testing technique – Each of the tests started by making necessary

TABLE II
SUMMARY OF TEST CONDITIONS

Test Solution Boiler Water	Test No.	Time Tubes Immersed in Boiler Water, Hr Operating[a]	Down-Time	Type Tubes	Tube Condition	Inhibitors[z]
b	1	3307	1656	E.R.W. ASTM-A-178 Grade A Boiler Tubes	New	None
b	2	3307	1656	"	New	No inhibition initially followed by inhibitor A.
b	3	3307	1656	"	New	Inhibitor B.
b	4	3338	1625	"	New	Inhibitor A.
c	5	3148	788	"	Same as Test #1	Continued in test with addition inhibitor A.
c	6	3148	788	"	New	No inhibition initially followed by inhibitor B.
c	7	3148	788	"	New	Inhibitor B.
c	8	3148	788	"	New	Inhibitor A.
c	9	2615	865	"	New	Inhibitor C.
c	10	2615	865	"	New	Inhibitor D.
c	11	2615	865	"	Same as Test #7	Inhibitor B.
c	12	2615	865	"	New	None-Tubes painted and paint spattered.
c	13	2548	1388	E.R.W. High strength, low alloy Steel No. 1	New	None.
c	14	2548	1388	E.R.W. ASTM-A-178 Grade A	New	No inhibition, copper tankless heating coil installed.
c	15	2548	1388	"	Same as Test #7	Inhibitor B.
c	16	2548	1388	"	New	No inhibition, physico-chemical water conditioner.
c	17	2354	674	E.R.W. High strength, low alloy Steel No. 2	New	None.

mechanical changes in the boilers (see tests 24 and 28, Table VII). New tubes required by the individual test (Table II) were installed, and the boilers were given a short boil-out with 0.3 per cent sodium hydroxide solution to remove residual oil. Following the boil-out, they were drained and rinsed with zeolite treated water. After the treated water rinse each boiler was rinsed with distilled water, drained and refilled with the base boiler water. The inhibitors, hardness constituents, etc., were added to the boiler water where required.

Then the boilers were started up. An operating log was maintained for routine maintenance and records of gas pressure, cooling water pressure, boiler water level,

Test Solution Boiler Water	Test No.	Time Tubes Immersed in Boiler Water, Hr Operating[a]	Down-Time	Type Tubes	Tube Condition	Inhibitors[g]
d	18	2354	674	E.R.W. ASTM-A-178 Grade A Boiler Tubes	New	Inhibitor A.
d	19	2354	674	"	New	Inhibitor B.
c	20	2354	674	"	New	Inhibitor B.
c	21	2849	1351	Copper bearing steel	New	None.
e	22	2849	1351	E.R.W. ASTM-A-178 Grade A Boiler Tubes	New	Inhibitor A.
e	23	2849	1351	"	New	Inhibitors B.
c	24	2845	1351	"	New	Inhibitor E.
c	25	2338	638	"	New	Inhibitor D.
f	26	2338	638	"	New	Inhibitor A.
c	27	2338	638	"	New	Inhibitor F.
c	28	2338	638	"	New	None-Baffle Plate installed 4 in. from Flue Sheet.
f	29	2850	798	"	New	Inhibitor B.
f	30	3980	2036	"	New	Inhibitor F.
c	31	2850	798	"	New	None-Copper flash plated tubes.
f	32	2850	798	"	New	Inhibitor G.

[a] Operating time refers to actual cyclic firing of the boilers as described elsewhere. Down-time is due to inspection and routine maintenance.
[b] Boiler water No. 1 was made up from air saturated distilled water.
[c] Boiler water No. 2 was made from air saturated distilled water and contained 100 ppm chloride.
[d] Boiler water 2 + 100 ppm calcium temporary hardness plus hardness chemical additions during tests.
[e] Boiler water 2 + 100 ppm temporary hardness + 100 ppm permanent hardness plus hardness chemical additions during test.
[f] Boiler water 2 + 500 ppm permanent hardness + 1000 ppm temporary hardness initially plus hardness chemical additions during test.
[g] Inhibitor code:
(A) Buffered chromate.
(C) Sodium molybdate.
(E) "Soluble oil".
(B) Borate-nitrate-nitrite.
(D) Sodium hydroxide.
(F) Sodium borate-sodium nitrite.
(G) Sodium nitrite without buffering.

and water analyses.

Samples of boiler water were taken twice weekly for analysis. Usual analyses consisted of pH, chloride, conductivity and inhibitor concentrations (when used). Periodically, the feedwater was analyzed for dissolved oxygen and evidences of carryover of boiler water salts to the condensate.

Each set of waterside conditions was tested for approximately six months. The test conditions and numbers are shown in Tables III through VII.

Water samples – Boiler water samples and feedwater samples were analyzed by the following methods:

pH measurements were made by electrometric techniques. Chlo-

TABLE III

SUMMARY OF THE RESULTS OF TESTS INVOLVING HIGH STRENGTH-LOW ALLOY STEEL TUBING UNDER TEST CONDITIONS SHOWN

Test No.	Material	pH Average[f]	Chloride as ppm Cl (Average)[f]	Dissolved Oxygen Concentration mls/liter (Average)[g]	Deepest Pit Measured	Number of Pits per Lineal Ft	Comments
1	Plain Carbon Steel	7.6	Not Added	5.2	0.015	3300	Generalized widespread pitting attack.
13	High strength—low alloy steel No. 1	10.0	95	5.1	0.003	615	Generalized surface brightening with slight attack.
17	High strength—low alloy steel No. 2	9.6	78	5.3	0.012	2028	Generalized pitting attack.
21	Copper bearing steel	7.7	87	4.5	0.028	2600	Generalized pitting corrosion.

[f] Boiler water [g] Feed water

ride analyses were made using the Mohr titration method (ASTM D-512-49).[4] Conductivity measurements were made with a L&N 4866 conductivity bridge and a glass "dip type" conductivity cell. Chromate was analyzed by the potassium-iodide-thiosulfate reaction in acid solution. Nitrate was determined by the ASTM D-992-52 method.[4] Nitrite was titrated by using a potassium permanganate-potassium iodide-sodium thiosulfate technique. Dissolved oxygen in the feedwater was measured by a modified Winkler test pattern after ASTM method D-888-49t.[4] The concentration of "soluble oil" in the boiler water was determined volumetrically with a Babcock Test Bottle using sulfuric acid to break the water-"soluble oil" emulsion.

Hardness analyses were made by means of the versenate titration. Copper in the boiler water was analyzed colorimetrically with Neo-Cuproine indicator. Borate was determined by the invert sugar-sodium hydroxide titration technique.

Corrosion measurement—A microscope calibrated for the purpose was used to measure corrosion pit depths and the number of pits per unit of metal surface were counted where possible.

Microscopic examination of the tubing in place in the boilers, difficult because of the curved surface of the tubes, was accomplished by taking a clay impression of the deepest visible pits and then measuring the height of the nodule formed on the clay, microscopically. Photographic evidence was also employed for comparison between tubes.

DISCUSSION

A summary of the results of the first 16 tests follows:

1—The buffered chromate type inhibitor is effective in halting and preventing corrosion caused by dissolved oxygen under test operating conditions. The minimum effective inhibitor concentration appears to be 2000 ppm.

2—The borate-nitrate-nitrite inhibitor is effective in halting and preventing corrosion caused by dissolved oxygen under these test conditions. The minimum concentration appears to be 3000 ppm. At this concentration, carryover is not observed in the test equipment.

3—Sodium molybdate and sodium hydroxide are not effective in halting dissolved oxygen attack in the test boilers.

4—The immersion of a copper tankless heating coil in the boiler water, out of contact with the tubes, does not accelerate or localize corrosive attack under these test conditions.

5—The physico-chemical feedwater conditioner used in this test program was ineffective in preventing or reducing corrosive attack under these test conditions.

6—A test with high strength, low-alloy steel tubing indicated a less severe attack on it than found on carbon steel tubing under the same test conditions. Pits were

TABLE IV

SUMMARY OF TEST RESULTS INVOLVING THE BUFFERED CHROMATE INHIBITOR

Test No.	Inhibitor Concentration ppm (Average)[g]	pH Average[g]	Chloride as ppm Cl (Average)[g]	Dissolved Oxygen Concentration mls/liter (Average)[h]	Deepest Pit Measured (In.)	Number of Pits per Lineal Ft	Comments
2	2116	10.5[a]	Not Added	5.4	0.010	840	Corrosion halted
4	1860	9.8	Not Added	5.3	0.010	25[c]	Slight corrosion[b]
5	1904	10.9	99	4.9	0,016	—[c]	Corrosion halted
8	2000	9.3	109	5.0	—	No Pitting	No corrosion
18	1939	11.3	78	5.3	—	—[d]	Only slight pitting attack
22	2184	10.9	79	4.5	—	—[e]	Only slight pitting attack
26	2007	10.1	85	5.6	—	—[f]	Only slight pitting attack

[a] Average pH after addition of inhibitor. Inhibitor added after corrosion was started.
[b] The slight pitting corrosion noted on the tubing in this test was attributed to an initially low chromate concentration.
[c] No increase was noted in the incidence of pitting after the addition of the proper concentration of inhibitor to the boiler water.
[d] 100 ppm calcium temporary hardness was added to the boiler water. Some slight incipient pitting was noted near the feedwater inlet. General condition of the tubes after test was good.
[e] $Ca(HCO_3)_2$, $CaSO_4$, & $MgSO_4$ was added initially at a low concentration.
[f] $Ca(HCO_3)_2$, $CaSO_4$, & $MgSO_4$ was added to the boiler water. The hardness concentration initially was very high. General condition of tubes after test was good.
[g] Boiler water.
[h] Feedwater.

Fig. 3a Test 17 – 2354 hrs oxygenated-distilled water. High strength low alloy steel No. 2. Generalized pitting corrosion (see Table III)

Fig. 3c Test 19 – 2354 hrs oxygenated-chloride-hardness containing boiler water (Table V). Borate-nitrate-nitrite inhibitor. No corrosion

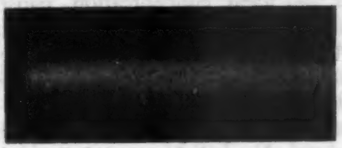


Fig. 3 Photographs of tubes after test

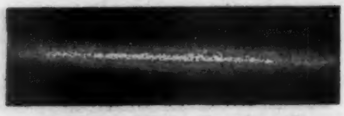


Fig. 3b Test 26 – 2238 hrs oxygenated-chloride-hardness containing boiler water (Table IV) chromate inhibitor. Only slight pitting. General tubing condition after test was good

Fig. 3d Test 21 – 2849 hrs copper bearing steel tubing in chloride-oxygenated distilled water. Generalized pitting corrosion (Table III)

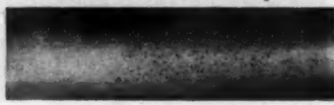
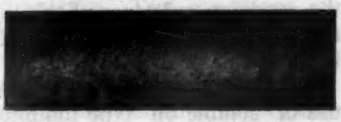


Fig. 3e Test 24 – 2845 hrs chloride-oxygenated-distilled water containing a "soluble oil" inhibitor. Localized corrosion (Table VI)



shallower and the incidence of pitting was lower.

A continuation of the investigation into high strength low-alloy steel tubes was made. Test 17, Fig. 3, shows the condition of alloy No. 2 tubes after 2354 hr operation and 654 hr down-time in chloride-oxygenated distilled water. As shown in Table III, the rate of attack was somewhat less than that of plain carbon steel, but considering the depth of pitting, the difference was not significant. The test-time with this tubing was also shorter than that of the carbon steel.

Tests 18, 22, and 26 (Table IV) were designed to test further the effectiveness of the SBI buffered chromate inhibitor in the presence of hardness constituents in the boiler water. Test 26, Fig. 3, shows a representative tube. This inhibitor is effective at the nominal concentration of 2200 ppm with moderate hardness in the boiler water.

Tests 19, 20, 23, and 29 (Table V) were designed to test the effec-

Fig. 4a Test 25 – 2338 hrs oxygenated-chloride-distilled water with the addition of sodium hydroxide to a pH of 11.0. Widespread corrosion (Table VI)

Fig. 4b Test 27 – 2338 hrs chloride-oxygenated-distilled water with the borate-nitrite inhibitor (Table VI) No corrosion

Fig. 4 Photographs after test

tiveness of the borate-nitrate-nitrite inhibitor at a nominal concentration of 3000 ppm. This inhibitor prevented the corrosion of new tubing in the presence of hardness in the boiler water. Test 19, Fig. 3, shows the tubing after tests under these conditions.

Test 21 (Tables II and III) shows the test conditions for copper bearing steel, which were comparable to those used for the other alloy steels. The results show that the corrosion resistance of this material does not differ significantly from that of plain carbon steel.

Corrosion prevention properties of "soluble oil" type inhibitors has been widely reported. These materials, strongly polar in nature, form an emulsion with water. They tend to prevent access of oxygen to the surface of metal. Therefore, when present in sufficient quantity in water, they are effective in preventing ferrous metal corrosion under various service conditions.

Test 24 (Tables II and VI) was to detect "soluble oil" in the boilers. A heavy foaming condition in the boiler water, the result of soluble oil-water emulsion at a nominal 2 per cent concentration, caused an immediate expected difficulty. In order to keep the boiler in operation and to maintain proper water level, it was necessary to install a small feed pump, controlled by the water level indicator. Oil in the gauge glass and a continuing requirement for "soluble oil" addition indicated the emulsion broke down. Fig. 3 shows the "soluble oil" inhibitor was not successful in preventing pitting attack of the boiler tubes.

Test 25 (Tables II and VI) involved further work on sodium hydroxide in the boiler water with new tubing. The previous tests with sodium hydroxide, on tubing, already corroded under uninhibited conditions were conducted to see if sodium hydroxide would stop corrosion. At a pH of 11.0, it was again ineffective, even with new tubing (Fig. 4). Sodium hydroxide can only reduce the rate of corrosion.

Tests 27 and 30 (Tables II and VI) involved the use of a sodium nitrite-sodium borate mixture. The

Fig. 4c Test 28 – 2338 hrs chloride-oxygenated-distilled water with baffle plate stagnation promoter installed in the boiler. Widespread rather than localized corrosion occurred (Table VIII)

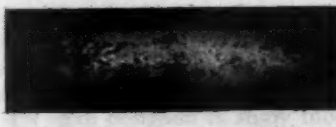


Fig. 4d Test 31 – 2850 hrs chloride-oxygenated-distilled water. Copper flash-plated carbon steel tubes. Wide-spread pitting corrosion (Table VII)

Fig. 4e Test 32 – 2850 hrs chloride-oxygenated-distilled water. Nitrite alone for inhibitor. No corrosion (Table VI)

nominal concentration of sodium nitrite was 2250 ppm; the nominal sodium borate concentration 250 ppm. Test 27, Fig. 4, shows the result. The inhibitor mixture was effective in preventing corrosion in chloride-oxygenated test water, both with and without adding a moderate concentration of hardness constituents to the water.

During the course of these investigations, considerable interest was directed toward the deep "isolated pit" type corrosion. In all tests, a generalized wide-spread pitting type corrosion occurred. A consensus grew that one cause of this serious localized corrosion, which had been observed in the field, might be localized stagnant areas in the boiler.

A mechanical change in one test boiler (Test 28) included a baffle plate 4½-in. from the flue sheet. A 1/16-in. clearance was provided between the baffle and the tubes and several ¾-in. drain holes were drilled in the baffle plate to facilitate filling, draining and to provide for boiler water-level equalization. The boiler water was chloride-oxygenated distilled water. No localization of corrosion occurred.

This corrosion, shown in Test 28 (Fig. 4), was the wide-spread pitting type. Test conditions are tabulated in Tables II and VII. If structural peculiarities in the boiler contribute to localized tube corrosion, the mechanical changes made here did not provide adequate simulation of these effects.

Frequently, tubing from operating boilers has revealed considerable metallic copper in the water-side deposits. This copper originates from corrosion of condensate return piping and is carried into the boiler with the feedwater.

Test 31 on copper flash-plated (acidic copper sulfate) tubes was to observe what effect they might have on corrosion. These tubes

TABLE V

SUMMARY OF TEST RESULTS INVOLVING THE BORATE-NITRATE-NITRITE INHIBITOR

Test No.	Inhibitor Concentration ppm (Average)[j]	pH Average[j]	Chloride as ppm Cl (Average)[j]	Dissolved Oxygen Concentration mls/liter (Average)[k]	Deepest Pit Measured (In.)	Number of Pits per Lineal Ft	Comments
2	7281	9.1	Not Added	—[a]	No pitting	—	No corrosion
6	7183	9.5[b]	100	5.1[a, c]	0.008	—[d]	Corrosion halted
7	6318	9.1	99	—[a]	No pitting	—	No corrosion
11	4000	9.1	92	—[a]	No pitting	—	No corrosion
15	2500	9.2	97	5.5[e]	No pitting	—	No corrosion
19	2694	9.4	119	5.5	No pitting	—	No corrosion[f]
20	2756	9.4	143	5.5	No pitting	—	No corrosion[g]
23	3214	9.4	86	4.6	No pitting	—	Slightly roughened tube surface[h]
29	3232	9.2	96	5.9	No pitting	—	No corrosion[i]

[a] Excessive carry-over experienced with the borate-nitrate-nitrite inhibitor at this concentration prevented the chemical determination of dissolved oxygen in the feedwater.
[b] Average pH value after the addition of the inhibitor. Corrosion was started in uninhibited boiler water.
[c] Dissolved oxygen concentration prior to addition of the inhibitor.
[d] Compared photographically with earlier inspections prior to inhibitor addition.
[e] Carryover was insignificant below the 3000 ppm concentration under these test conditions.
[f] Initially 100 ppm calcium temporary hardness ($Ca(HCO_3)_2$) was added to the boiler. Additional $Ca(HCO_3)_2$ was added during the test period.
[g] Test made with new tubing. Test numbers 7, 11, and 15 were made without replacing tubing.
[h] 100 ppm of both permanent and temporary hardness was added to the boiler water initially. Additional hardness was added during test.
[i] 1000 ppm temporary harness was added to boiler water initially along with permanent hardness. Hardness additions were made during the test.
[j] Boiler water.
[k] Feedwater.

were installed in one test boiler and operated for 2850 hrs during which 798 hrs of down-time occurred (Tables II and VII). Results showed no significant increase in corrosion as a result of copper deposition.

Test 32 involved the use of only sodium nitrite as an inhibitor in chloride-oxygenated-distilled water. No corrosion occurred during this test (Fig. 4, Tables II and VI). Note, however, that the average pH was 9.9 and the lowest pH recorded was 8.7. If the pH had fallen below 7, the inhibitor would not have been effective. Sodium borate is normally added to this inhibitor to insure maintenance of an alkaline pH. Therefore, inclusion of sodium borate in the sodium nitrite inhibitor formulations is recommended.

WET STORAGE TESTS

During the accelerated tube corrosion test program, there was considerable discussion of summer storage periods of boilers. It was felt that many of the tube corrosion difficulties begin at that time. Tests were designed to study tube corrosion during boiler storage periods.

Test Equipment—Small steel boxes were constructed from ¼-in. carbon steel plate (Fig. 5). One box was designed so that deaeration could be carried out, and the watersides were blanketed and sealed under nitrogen (Fig. 5). A sealed glass plate was provided for test observation.

Two sides of the carbon steel boxes served as tube sheets so that tubes could be "rolled in" simulating the mechanics of boiler construction.

Boiler tubes — Electric resistance welded tubes made to ASTM-A-178 Grade A specifications were used in these tests. Typical tube metal analyses for these tests are shown in Table VIII.

Fig. 5-A Storage container for atmospheric tests

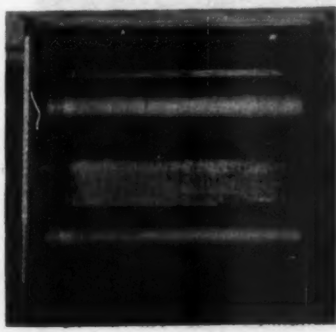


Fig. 5-B Storage container for the test under deaerated (air free) water

TABLE VI

SUMMARY OF TEST RESULTS INVOLVING THE SODIUM HYDROXIDE, SOLUBLE OIL, SODIUM MOLYBDATE, SODIUM BORATE-SODIUM NITRITE AND SODIUM NITRITE

Inhibitor	Test No.	Inhibitor Concentration ppm (Average)[g]	pH[g]	Chloride as ppm Cl (Average)[g]	Dissolved Oxygen Concentration mls/liter[h]	Deepest Pit Measured	No. of Pits per Lineal ft of tubing	Comments
Sodium molybdate	9	1978	10.5[a]	98	4.5	0.005	—	Some inhibition of continued corrosion
Sodium hydroxide	10	—[b]	11.0[a]	90	4.7	0.014	—[c]	No inhibition of further corrosion
Sodium hydroxide	25	—[b]	11.0[d]	90	4.7	—	—	General corrosion
Soluble oil	24	1.4% (by volume)	8.4	93	3.9[e]	—	—	Localized corrosion where film broke down
Sodium borate—Sodium nitrite	27	$Na_2B_4O_7$ 276 $NaNO_2$ 2079	9.7	76	5.1	—	—	No corrosion
Sodium borate—Sodium nitrite	30	$Na_2B_4O_7$ 2030[f] $NaNO_2$ 332	9.6	90	5.1	—	—	No corrosion
Sodium nitrite	32	2276	9.9	100	6.1	—	—	No corrosion

[a] Average pH value after addition of the respective inhibitors. Corrosion initiated in chloride-oxygenated distilled water.
[b] The addition rate of sodium hydroxide was controlled by boiler water pH.
[c] Deep pits occurred which were overlayed by large carbuncles.
[d] Sodium hydroxide was added at start of test with new tubing in this case.
[e] Carryover of soluble oil into condensate resulted in erratic dissolved oxygen determination. Serious foaming occurred in boiler water.
[f] A high level of temporary ($Ca(HCO_3)_2$) hardness and permanent ($CaSO_4$, $MgSO_4$) was added after 1650 hrs of operation.
[g] Boiler water.
[h] Feedwater.

Testing technique—The inside surfaces of the test boxes were carefully cleaned and sections of new tubing were rolled into the simulated tube sheets. All surfaces in contact with water were then cleaned with trichloroethylene, rinsed with distilled water and dried with methyl alcohol.

Test boxes were filled with distilled water; sodium chloride and sodium sulfate were added as an electrolyte, depending on proposed test conditions, as was the proper chemical inhibitor.

They were stored at room temperature for periods of one to three months. A visual inspection was made each day to observe the presence or absence of corrosion. The water in the test boxes was analyzed at the beginning and at the end of each test, by methods described in section one. The extent of corrosion was recorded photographically.

DISCUSSION

Fig. 6 shows a comparison between tubes immersed in chloride-containing water under atmospheric and deaerated (air boiled outlayed up under nitrogen seal) conditions. Table IX shows the initial and post test analysis of the water from each test. The effect of oxygen dissolved from the atmosphere above the test water is readily apparent in the widespread pitting attack on the tubes of the box open to air. No preferential corrosion locations were noted on the tubing. The corrosion rate lessened

Fig. 6 Corrosion by dissolved oxygen

Fig. 6a Post test condition of tube stored in distilled water containing chloride and open to the atmosphere (2713 hrs) Test 1 (Table IX)

Fig. 6b Post test condition of tube stored in deaerated distilled water containing chloride for 2713 hrs. Test 2 (Table IX)

TABLE VII

SUMMARY OF TEST RESULTS INVOLVING THE EFFECT OF MECHANICAL CHANGES ON TUBE COMPOSITION

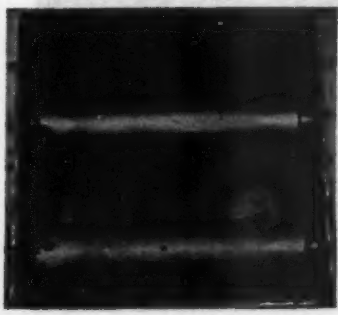
Test No.	Conditions	pH Average	Chloride as ppm Cl (Average)	Dissolved Oxygen Concentration mls/liter (Average)	Deepest Pit Measured	Number of Pits per Lineal Ft	Comments
12	Paint spattered tubes	7.6	102	4.8	Etched appearance	—	Corrosion accelerated locally.
14	Copper heating coil immersed in boiler water.	10.4	89	5.3	0.011	2770	Did not appear to localize or accelerate corrosion.
16	Physico-chemico water conditioning device.	10.1	94	4.8	0.020	2940	The device did not decrease corrosion.
28	Baffle plate installed adjacent to flue sheet.	10.5	88	5.1	0.016	1272	No localization of corrosion.
31	Tubes flash plated with copper.	9.0	93	6.0	0.018	3423	No definite acceleration of corrosion could be established.

Fig. 7 Localized corroded areas on tube removed from chloride containing distilled water after 2328 hrs. Water was inhibited with 2000 ppm SBI chromate. Carbuncles have been removed

in time, due to the formation of a film of hydrated iron oxide on the surface of the water which restricted oxygen access to the water. Further proof was the high concentration of ferrous iron in the water after the test. This iron oxidized and precipitated when exposed to air.

After demonstrating the effect of dissolved oxygen on steel in this manner, researchers tested, under storage conditions, the efficiency of the inhibitors which had shown promise.

Fig. 8 Large carbuncles on tubes and tube sheets in a chloride containing (140 ppm) Chromate treated (2000 ppm). Storage test within 48 hrs. Test 13 (Table IV)



New tubing was installed in the test boxes, and after cleaning, new chloride-containing test solutions were added. SBI chromate at a concentration of 2000 ppm was added to the water in one test box while 3000 ppm of the proprietary borate-nitrate-nitrite inhibitor was added to the other.

Incipient corrosion was visible within 24 hr in the chromate treated storage box while the tubes in the borate-nitrate-nitrite treated test did not corrode. Fig. 7 shows the corroded condition of the tubes from the two boxes after 3 months test time. The large localized pitted areas located on the tubes from the chromate treated storage box were covered with large carbuncles before cleaning. When analyzed spectrographically, the material in the carbuncle consisted of iron and chromium, presumably in the oxide or hydrated oxide form. The material was not crystallized well enough for identification by X-ray diffraction. Table IX shows the water analyses made for each test. Little evaporation took

place in the chromate treated box since chemical concentrations remained approximately the same.

Tests investigating the variables associated with the preceding, attempting to find the cause of localized attack in the chromate treated storage test, included studies of stray current effects associated with a nearby motor-generator set, the addition of another salt (Na_2SO_4) for buffering effect, the initial boiling after inhibitor addition to promote protective film formation. The tests showed that none of these variables had any effect on localized corrosion.

The remaining variable was the chloride concentration in the test water. Another series of tests was made using varying chloride concentrations: 34 ppm for test box No. 1, 73 ppm for test box No. 2, and 140 ppm for test box No. 3. Large carbuncles formed on the tubes and tube-tube sheet junctions of the test box containing 140 ppm chloride within 48 hr. (Fig. 8)

Literature[5] indicated that the sulfate concentration would have the same effect on the sodium nitrite inhibitor as chloride on the chromate inhibitor. Tests to determine the concentrations of sulfate which would reduce the effectiveness of a sodium borate-sodium nitrite mixture at a nominal concentration of 2500 ppm (Table IX) were devised.

The results showed that the sulfate ion would interfere with the inhibitive properties of the sodium nitrite inhibitor. More than 500 ppm of sulfate concentrations were required to produce corro-

Fig. 9 Severe corrosion on tubes stored in sulfate containing (705 ppm) Borate-Nitrate treated (2510 ppm). Test 18 (Table IV)

sion at nominal inhibitor concentration (2500 ppm). The type of attack differed from that of chloride on the chromate inhibitor (Fig. 9). In the case of the sulfate interference with the nitrite inhibitor properties, the attack was more general than localized.

Further testing with the borate-nitrate-nitrite inhibitor (nominal 3000 ppm concentration, Test 20-22) showed a similar corrosion occurring at approximately 200 ppm concentration of sulfate ion ($SO_4=$). Since sodium nitrite makes up approximately 30% of this inhibitor compound, it appears that if the sodium nitrite concentration is less than four times the sulfate concentration, proper storage inhibition is not attained under these test conditions.

CONCLUSIONS

Tests conducted since April, 1957, have resulted in the following conclusions:

1 — The Steel Boiler Institute buffered chromate inhibitor at a nominal concentration of 2200 ppm is satisfactory in preventing dissolved oxygen corrosion of plain carbon steel tubing when hardness in the boiler water is present.

2 — Both the proprietory borate-nitrate-nitrite inhibitor at a concentration of 3000 ppm and the borate-nitrite (10% sodium borate, 90% sodium nitrite) mixture (at 2500 ppm) are also satisfactory in preventing dissolved oxygen attack on plain carbon steel tubing. These inhibitors were tested with and without hardness in the boiler water.

3 — The "soluble oil" inhibitor was not a satisfactory inhibitor under test conditions. Considerable pitting and corrosion of the tubing as well as a serious foaming condition, resulting in boiler water carryover into the stream, occurred. Under boiling conditions the soluble oil emulsion partly broke down,

TABLE VIII

TUBE METAL ANALYSIS—STORAGE TESTS

Description	Tubes Used in the Oxygen Corrosion Demonstration Test	Tubes Used in Chromate and Borate-Nitrate Nitrite Test	Tubes Used in Evaluating Chromate-Chloride Relationship
Carbon, as C	0.12	0.10	0.13
Silicon, as Si	<0.01	0.01	<0.01
Manganese, as Mn	0.40	0.45	0.45
Phorpherus, as P	0.014	0.012	0.010
Sulfur, as S	0.046	0.032	0.029
Nickel, as Ni	0.045*	0.10*	0.04*
Chromium, as Cr	Not Detected	0.05*	0.023*
Vanadium, as V	"	—	0.012*
Molybdenum, as Mo	0.008*	0.015*	0.010*
Copper, as Cu	0.055*	0.037*	0.076*
Aluminum, as Al	0.003	0.014*	0.007*

* Spectrographic analyses

TABLE IX
WET STORAGE TESTS

Test No.	Test Conditions	Length of Test Hours	Inhibitor**	Inhibitor Conc. ppm Initial	Final*
1	Test box stored open to atmosphere.	2713	None	—	—
2	Deaerated and stored under nitrogen.	2713	None	—	—
3	Test box stored open to atmospheric conditions.	2328	A	2000	1977
4	Same as 3.	2328	B	3000	4350
5	Boiled and allowed to cool to room temperature. Stored open to atmosphere.	1656	A	2000	2644
6	Grounded by copper wire to nearby motor generator set. Stored open to atmospheric conditions.	1656	A	2000	2059
7	Same as 3.	1656	A	2000	1983
8	Same as 3.	2568	A	2000	2682
9	Same as 3.	2568	A	2000	3369
10	Same as 3.	2568	A	2000	4100
11	Same as 3.	1320	A	2308	2721
12	Same as 3.	1320	A	2472	3440
13[2]	Same as 3.	1320	A	2103	3660
14	Same as 3.	768	C	2490	2499
15	Same as 3.	768	C	2531	2620
16	Same as 3.	768	C	2508	2600
17	Same as 3.	1632	C	2728	2650
18	Same as 3.	1632	C	2510	2575
19	Same as 3.	1632	C	2518	2669
20	Same as 3.	168	B	3125	—
21	Same as 3.	168	B	2995	—
22	Same as 3.	168	B	3400	—

* Concentration of chemicals due to natural evaporation of water in test boxes.
** Inhibitor A-SBI chromate.
Inhibitor B-Borate-nitrate-nitrite.
Inhibitor C-Borate-nitrite.

[1] No determination due to large amounts of dissolved and precipitated iron. Solution was reducing due to dissolved ferrous iron.

[2] Water exposed to laboratory atmosphere and therefore assumed to be saturated (6 mls/liter O_2) room temperature.

as exhibited by oil in the boiler gauge glass.

4 – A confirmation test, with sodium hydroxide as an inhibitor in oxygenated-distilled boiler water containing sodium chloride, showed the ineffectiveness of this chemical compound (alone) in new plain carbon steel tubing.

5 – An additional "high strength, low alloy" tube material and a copper bearing steel were tested in oxygenated-distilled boiler water containing sodium chloride.

Chloride ppm Initial	Chloride ppm Final*	Sulfate ppm Initial	Sulfate ppm Final*	Dissolved Oxygen mls/liter	pH	Comments
	216	None	—	(1)	6.6	Severe corrosion.
	214	None	—	0.07	9.2	No corrosion.
324	336	None	—	(2)	9.5	Large carbuncles. Pit corrosion under carbuncles.
306	340	None	—	(2)	9.2	No corrosion.
300	348	None	—	(2)	9.2	Same results as Test 3.
280	284	None	—	(2)	9.4	See Test 3.
254	260	None	—	(2)	9.2	See Test 3.
None	None	None	—	(2)	8.0	No corrosion.
128	172	None	—	(2)	8.6	See Test 3.
240	346	None	—	(2)	8.6	See Test 3.
29	34	None	—	(2)	8.1	No corrosion.
62	73	None	—	(2)	8.1	No corrosion.
114	134	115	140	(2)	8.4	Large carbuncles formed within 48 hr of test start.
None	None	60	55	(2)		No corrosion.
None	None	140	150	(2)		No corrosion.
None	None	275	230	(2)		Possible incipient corrosion.
None	None	540	575	(2)		No corrosion.
None	None	705	705	(2)		Severely corroded.
None	None	980	1050	(2)		Severely corroded.
—	—	525	—	(2)		Severe corrosion.
—	—	260	—	(2)		Slight corrosion.
—	—	175	—	(2)		No corrosion.

Neither alloy (Table II and III) had markedly less pitting than plain carbon steel. It appears that low alloy steels of this type will not provide a satisfactory solution to the tube corrosion problem.

6 – Copper flash-plating of plain carbon steel boiler tubes with acidic copper sulfate, and its subsequent operation in oxygenated distilled boiler water containing sodium chloride, did not lead to accelerated corrosion. Copper in the form of copper sulfate was added to the boiler water, periodically.

7 – Sodium nitrite, alone, was an effective inhibitor in oxygenated-chloride containing boiler water. However, sodium nitrite should not be used without a

chemical buffer to produce an alkaline pH.

8 – In tests reported here, the Steel Boiler Institute buffered chromate inhibitor at a nominal 2200 ppm concentration was a satisfactory wet storage inhibitor under atmospheric conditions, provided that the chloride concentration in the water was not more than 100 ppm. Storage tests were conducted with chloride concentrations only 73 ppm and 140 ppm; however, the accelerated test boilers, using the SBI chromate compound, were operated at the 100 ppm chloride (nominal) level, and down-time storage during test periods occurred without adverse effects. Most original fill waters will be below this chloride figure and with no, or quite small, losses of steam or water from the system, thus avoiding concentration of boiler water salts. However, this difficulty with the SBI chromate should not be a serious problem.

9 – Borate-nitrite at a nominal 2500 ppm and proprietary borate-nitrate-nitrite at 3000 ppm were satisfactory atmospheric wet storage inhibitors. Chloride at concentrations to 300 ppm does not interfere, but high sulfate concentrations in the water would interfere with the effectiveness of these inhibitors under the test conditions used. A sulfate concentration of at least 500 ppm at the nominal borate-nitrite inhibitor concentration of 2500 ppm was necessary. Natural waters containing this high concentration of sulfate are unusual. However, it was found that the nitrite and sulfate concentrations were related. The sodium nitrite concentration should be at least four times the sulfate concentration for storage protection, emphasizing the importance of low system losses.

Recommendations – If adhered to, the following should produce a general increase in the operating reliability of low-pressure steam (15 psig max) boilers.

1 – The watersides of all new boilers should be cleaned by boiling it out with a 0.5% solution (approx. 4 oz/5 gal water) of soda ash (Na_2CO_3 n H_2O), then by draining and rinsing it with fresh water.

2 – The boiler should be refilled with water and the proper chemical inhibitor added. It should then be fired for a short period to insure proper dissolution and mixture of the inhibitor in the water. Never permit a boiler to stand in wet storage without a proper chemical inhibitor in the boiler water. Choose an inhibitor from the following:

Steel Boiler Institute chromate inhibitor at a concentration of 1.5 oz per five gal of water. If the boiler water contains more than 100 ppm (approx. 6 grains/gal) chloride, wet storage tests show that severe localized corrosion occurs.*

A borate-nitrate-nitrite inhibitor (approx. 40% sodium borate (borax), 30% sodium nitrite and 30% sodium nitrate) at a concentration of 2 oz per five gal of water. *If the boiler water contains

*Refer to conclusions numbers 8 and 9 in preceding text.

less than four times as much sodium nitrite as sulfate, wet storage tests show that corrosion occurs. (This would be equivalent to approx. 30 grains/gal for the borate-nitrite or 12 grains/gal for borate-nitrate-nitrite inhibitor).

A borate-nitrite mixture (90% sodium nitrite and 10% sodium borate) at a concentration of 1.7 oz per five gal of water. Also the note concerning sulfate concentration effects applies to this inhibitor.

3 – It is recommended that prior to summer or other wet-storage periods the boiler be drained and refilled, in view of the possible concentration of undesirable anions, such as sulfate and chloride in the boiler water due to make-up for steam losses and system leaks during the heating season. After draining and then refilling with fresh water, the proper concentration of the desired inhibitor should be added and the boiler fired for a short period to insure proper solution and mixture of the inhibitor in the boiler water before leaving out of service.

During this short firing period, air must be steamed out of the system through the vents and the boiler then shut-down and sealed for storage. The boiler must not be allowed to stand after draining and refilling, unless it's fired to drive off dissolved oxygen.

4 – Tests show that these inhibitors. when used in specified amounts, stopped corrosion which was already in progress. But due to the uncertainty of what amount and type of waterside deposit accumulations may be present in old units, depending on local conditions, cleaning these boilers before adding the proper chemical inhibitor is recommended. Chemical cleaning agents and inhibitors can be used. (The boiler owner should contact his local plumbing and heating service.)

ACKNOWLEDGMENTS

This research was carried out at the Research Center of The Babcock & Wilcox Co. Tubular Products Division, for the Steel Boiler Institute. The guidance of the Engineering Committee of the SBI is gratefully acknowledged. Appreciation is also extended for the help provided the writers by H. F. Hinst, Chief Metallurgist, Keystone Plant of the Tubular Products Division, Babcock & Wilcox Co.

BIBLIOGRAPHY

(1) Keilbaugh, W. A. and Pocock, F. J. Corrosion Inhibition on Tubes in Low Pressure Steel Boilers, *Heating, Piping and Air Conditioning*, February, 1958.

(2) Butler, G. M. A., Ison, H. C. K., Model Boiler Tests on the Influence of the Copper Content of Steel on the Corrosion of Tubes in Artificial Sea Water. *The Institute of Marine Engineers*. Feb. 12, 1957.

(3) Robertson, W. D., *Molybdate and Tungstate as Corrosion Inhibitors and the Mechanisms of Inhibition*. Journal of The Electrochemical Society, Vol. 98 No. 3, 1951.

(4) Manual on Industrial Water, ASTM Committee D-19, ASTM Special Publication No. 148, American Society of Testing Materials, Phila., Penna.

(5) Sussman, Sidney, et al. Experiences with Sodium Nitrite Unpredictable Corrosion Inhibitor, Industrial and Engineering Chemistry, Vol. 51, No. 4, pp. 581-584, April, 1959.

DISCUSSION

John L. Fuller, Cleveland, Ohio: What effect is noted from too heavy a concentration of the inhibitors?

Author Keilbaugh: One problem would be that of solubility. For steam boilers operating up to 15 psig at saturation temperatures, inhibitor concentrations of up to 3000 ppm are not a problem. However, when using anodic inhibitors at high concentrations, a high rate of heat transfer is not recommended. With high rates of heat transfer, waterside deposits could result from the necessary high boiler water salt concentration.

Another problem of high inhibitor concentrations is that boiler water foaming can occur. Our experience with the test boilers indicates that inhibitor concentrations below 3000 ppm are tolerable.

W. B. Morrison, Portland, Ore.: Has any study been made in relation to water with a silicon treatment? This is a problem that prevails in our area. Have any tests been run to determine what effect the inhibitors might have or if they would be recommended in connection with water having a silicon content?

Author Keilbaugh: Is this problem with high silicon boiler waters one of the formation of silicate scales on the surface of the heat transfer tubes? These scales are difficult to remove.

Author Pocock: Perhaps you are referring to a slightly higher duty boiler than we are. Is this at pressures as low as 15 psi?

W. B. Morrison: Some higher pressure boilers have been worked up to 400 psi. Complete water treatment is carried on, yet over a period of years a build up of silicon scales, even in low pressure boilers, often appears. This is not a considerable factor in low pressure boilers, but is there a difference in reaction with silicon?

Author Keilbaugh: We were not aware that silica is a problem in low pressure boilers (below 15 psig). It might be advisable for manufacturers of proprietary compounds to consider this, since siliceous chemicals are sometimes intentionally added to the inhibitor formulations. There is a difference in the reaction of silica in boiler water dependent on heat and pressure. Analcite scale (sodium aluminum silicate) is known, for instance, to form in a closed system above the temperature of 350 F saturated.

Robert Krueger, Des Plaines, Ill.: Have any tests been carried out where the chromate concentration of the inhibitor was held constant, but where the pH of the solution was varied? What is the proper pH, if there is one?

Author Keilbaugh: In connection with the proprietary chromate inhibitor used, the pH was controlled between 9 and 10 by the addition of a buffering chemical in the inhibitor formulation. If a buffer were not added, under some conditions the pH might decrease to the acid side and a problem would arise in that manner. Our experience indicates that a pH of 9-10 is to be recommended.

1735

No. 1735

Design of Wet Cell Air Humidifiers

MELVIN W. FIRST

Wet cell washers have been used for many years for cooling, humidifying, dehumidifying, and cleaning ventilation and process air. Manufacturers are able often to supply satisfactory performance data for their own equipment, but little information of a systematic nature has been published relating the nature and amount of cell packing, nozzle design, or air and water rates to air washer performance.

This report is concerned exclusively with the performance of wet cell washers when employed for adiabatic cooling and humidification of outside air, but these data also have a high degree of relevance to the other heat and mass transfer processes for which this equipment may be employed.

The recent availability of a large variety of synthetic materials which offer interesting characteristics as wet cell packings (plus novel forms of older materials such as glass and metals) has expanded greatly the choices available to the designer of this type of equipment and, doubtless, has increased the need for improved performance data on a much wider variety of materials.

THEORETICAL CONSIDERATIONS

Air humidifiers, in which untempered water is recirculated from a sump having a total capacity many times the minute volume water rate, have the characteristics of an adiabatic system. Heat transferred to the water from the air tends to heat it, but evaporation tends to cool it, and when the unevaporated water is recirculated, it attains the wet bulb temperature of the air; at this point these two effects balance and the water temperature thereafter remains unchanged.

In practice, this perfect balance is seldom obtained, e.g. heat may be gained or lost through the washer cabinet from the surround-

Melvin W. First is a consulting and research engineer, Newton Highlands, Mass. This paper was prepared for presentation at the ASHRAE Semiannual Meeting in Chicago, February 13-16, 1961. The data presented here were obtained in experiments conducted for the Air Cleaning Div. of the Buffalo Forge Co.

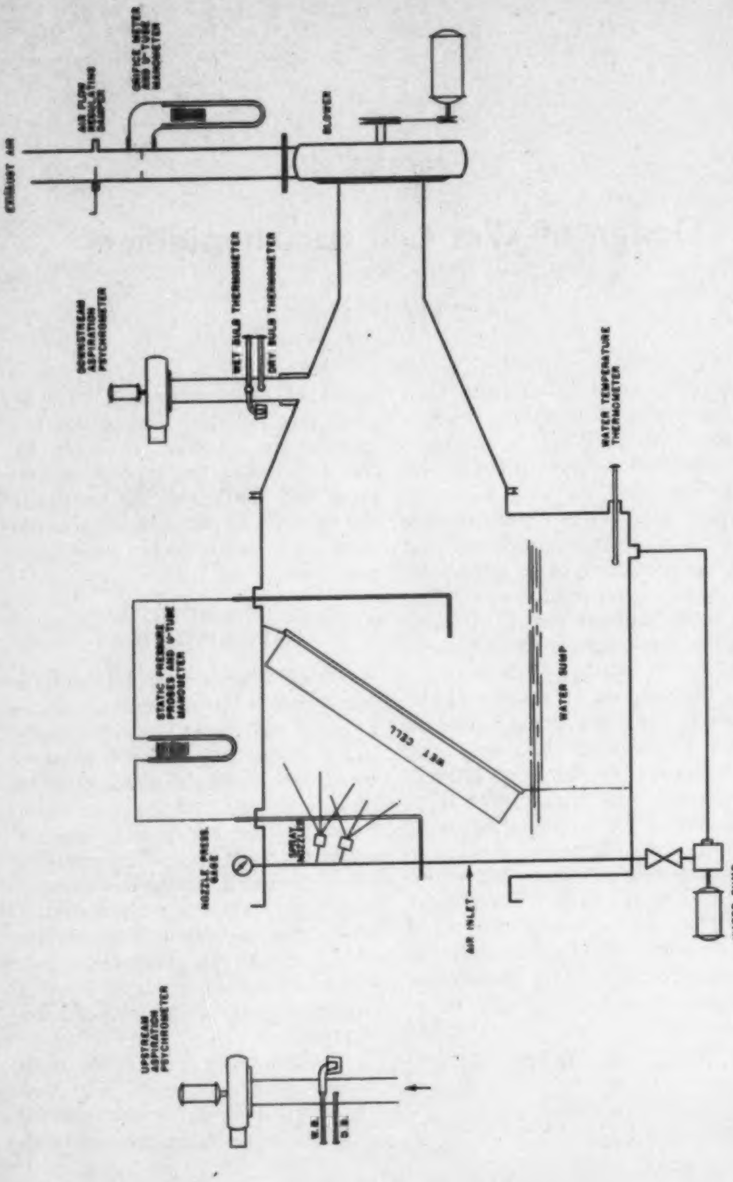


FIG. 1 ARRANGEMENT OF EXPERIMENTAL APPARATUS

ing air, or the temperature of the make-up water may be above or below the wet bulb temperature. Except when the water is artificially heated or cooled, however, the observed effects are sufficiently close to those obtainable from adiabatic humidifiers that it is generally permissible to ignore the deviations noted above.

In any case, the larger the washer the less a factor extraneous heat sources become in overall performance. This is because the ratio of exposed surface to cubical contents decreases as the size increases, so that, although heat gains or losses per sq ft of cabinet remain unchanged, the heat gain per cu ft of air flow decreases rapidly when capacity exceeds a few thousand cfm of air.

On the other hand, with small test units of 100-500 cfm capacity, heat exchanges through the washer cabinet and temperature differences of make-up water are often significant. Precautions must be taken to minimize deviations from a true adiabatic process if the collected data are to be meaningful.

To develop theoretical equations relating humidification performance to the physical characteristics of the apparatus, it is necessary to consider the processes that take place at the boundary between the air stream and the water film. First, water is evaporated from the liquid surface and finds its way into the air stream, increasing its absolute humidity; second, heat losses or gains from the bulk liquid are balanced by corresponding sensible and latent heat changes in the air stream. These relationships can be quantitated by a material and heat balance as follows:

$$-dL = -wdH \qquad (1)$$

$$-L(dT) = -ws(dt) - wr(dH) \qquad (2)$$

where dL = mass rate of water evaporation (lbs per hr)
w = mass air flow rate (lbs of air per hr)
dH = absolute humidity increase of air (lbs of water vapor per lb of dry air)
dT = change in temperature of the water (F)
s = humid heat of air = 0.24+0.45H (Btu/(F) (lb dry air)
dt = change in dry bulb temperature of air (F)
r = latent heat of vaporization (Btu/lb water)

Since the bulk water temperature remains unchanged at the wet bulb temperature of the air in an adiabatic humidifier, LdT = 0 for this special case and

$$-ws\,(dt) = wr\,(dH) \qquad (3)$$

This indicates that the change in the sensible heat content of the air stream is equal to the latent heat gain but opposite in sign; i.e. total heat content of the air stream remains unchanged and the heat required to evaporate additional moisture is obtained from lowering the temperature of the dry air and moisture already evaporated.

Determination of the rate of transfer of sensible heat between (1) the main body of air at temperature, t, and (2) the water film at temperature, t_1, requires the introduction of a rate coefficient, h, equal to the number of Btu transferred per hr per sq ft of inter-

face between air stream and liquid and per deg temperature difference between air stream and water film, as follows:

$$-ws\,(dt) = h\,(dA)\,(t - t_i) \qquad (4)$$

where dA = the differential area of contact between air stream and water film.

Similarly, the rate of diffusion of water vapor between the air at the interface (having an absolute humidity = H_i) and the main body of the air stream (having an absolute humidity = H) is given by the following mass transfer equation:

$$-w\,(dH) = k\,(dA)\,(H - H_i) \qquad (5)$$

where k = coefficient of vapor diffusion in lb of vapor per hr per sq ft of interface between air stream and liquid and per unit humidity difference between gas and interface. Temperature of the water film, t_i, is substantially constant and equal to the wet bulb temperature since the two phases are in equilibrium at the interface. Similarly, humidity of the film of air at the interface, H_i, is constant and equal to saturation at the wet bulb temperature (H_w). Making these substitutions and integrating equations 4 and 5 gives:

$$\ln \frac{t_1 - t_w}{t_2 - t_w} = \frac{hA}{ws} \qquad (4a)$$

and

$$\ln \frac{H_w - H_1}{H_w - H_2} = \frac{kA}{w} \qquad (5a)$$

where the subscripts "1" and "2" refer to entering and leaving conditions, respectively.

The ratio of the humidity difference between air at exit and saturation humidity at the wet bulb temperature, and the humidity difference between air at entry and saturated air may be obtained from equation 5a by determining the anti-log and inverting. This gives:

$$\frac{H_{w1} - H_2}{H_{w2} - H_1} = \frac{1}{e^{\frac{kA}{w}}} = e^{\frac{-kA}{w}} \qquad (5b)$$

The wet bulb temperatures of entering and leaving air are substantially equal in an adiabatic humidification unit so that $H_{w1} = H_{w2}$ and E, the fractional humidification efficiency, i.e. actual humidification divided by that obtainable in an infinite apparatus, equals:

$$1 - \frac{H_w - H_2}{H_w - H_1} = 1 - e^{\frac{-kA}{w}} \qquad (5c)$$

or

$$\frac{H_2 - H_1}{H_w - H_1} = 1 - e^{\frac{-kA}{w}} \qquad (5d)$$

Rate coefficients "h" and "k" of equations 4a and 5a are interrelated, so that these two equations may be used interchangeably. Since it is more convenient, experimentally, to measure wet and dry bulb temperatures, equation 4a is preferred and it takes a form similar to that of equation 5d, as follows:

$$E = \frac{t_1 - t_2}{t_1 - tw} = 1 - e^{\frac{-hA}{ws}} \qquad (4b)$$

The interfacial area (A) of equations 4b and 5d refers to the total effective area of contact be-

tween liquid and gas phases and not to the total surface of the packing material, since all the packing material is never completely wetted and, in addition, a certain portion of the surface is made inactive by areas of contact between packing elements. In practice, it is almost never possible to determine the true interfacial area of contact in a wetted fiber cell or device of similar nature. It is generally helpful, however, to consider the total effective interfacial area, "A," as the product of "a," the interfacial area per unit volume of cell, "b," the cross sectional area of the cell, and "z," the cell depth. In equation 4b, air flow rate, w, is in terms of lbs of dry air per hr, but air flow may be considered on a volumetric basis by substituting $60\rho Q$ for w, where Q = cfm and ρ = lb per cu ft of air. The volumetric flow rate is also equal to the product of air velocity "V" and cell face area "b." Making these substitutions in equation 4b gives

$$E = \frac{t_1 - t_2}{t_1 - t_w} = 1 - e^{\frac{-(hA)\ (z)\quad 1}{(V)\quad 60\rho s}} \qquad (4c)$$

Since density of air differs little over the temperature range encountered in supply air humidification problems, "ρ" may be considered as constant with little error. Similarly, over the temperature and humidity range encountered, changes in the value of the humid heat "s" will seldom, if ever, exceed 5%; this term, also, may be considered as a constant. Under these conditions, equation 4c indicates that humidification efficiency increases with depth of cell and with decrease of air flow rate. Efficiency also increases with effective interfacial area per unit volume of cell, and with greater heat transfer coefficient. For fiber cells, the unit effective interfacial area "a" may be augmented by utilizing finer fibers, packing greater weights of fibers into the cell and by more effective fiber wetting through the use of special geometric fiber arrangements and by alterations of the spray pattern and mean spray droplet size.

Transfer coefficient "h" is not constant but is thought to be a complex function of a number of variables. The most important of these are the shape of the solid surface elements and the velocity of air past the wetted surface. In all instances but one, the packing materials investigated in this study were cylindrical (or nearly so), so that in this case the principal variation in the value of the transfer coefficients was related to air velocity through the packed bed.

EXPERIMENTAL EQUIPMENT AND PROCEDURES

Fig. 1 illustrates, diagramatically, the arrangement of apparatus used in the humidification tests. A one in. thick insulating glass fiber blanket (not shown) sheathed the entire washer cabinet and water piping as it was found, experimentally, that heat exchanges through the walls were influencing results significantly when a large difference existed between dry and wet bulb temperature of the ambient air.

Air flow rates were measured with a 5⅝ x 4 in. orifice meter and alcohol-filled U-tube manometer placed on the discharge side of the air blower. The meter was calibrated by standard Pitot-static tube traverses at 10 air flow rates, representing a velocity range through the humidification cell of 100-700 fpm. Water flow rates were measured by noting water pressure at entry to the nozzle and referring to "pressure-delivery" curves for each nozzle.

Inlet and outlet dry- and wet-bulb temperatures, and temperature of the circulating water, were measured with calibrated psychrometric mercury thermometers having a stem length of 10 in. and a temperature range of 30-120 F in ½ deg divisions. An aspiration psychrometer with the air inlet located just above the center of the washer inlet (see Fig. 1) was used to measure inlet conditions. Heat exchanges with the surroundings by conduction and radiation were negligible since the temperature of the air entering the aspiration psychrometer was, for all practical purposes, identical with that en-

Table I—Wet Cells

Cell	Description	Thickness in.	Packing Surface sq ft/ cu ft	Packing Surface total sq ft
Crimped Metal Ribbon	Packed with herringbone-crimped aluminum ribbons 0.009 in. thick, 12 in. long, and 2,4, or 6 in. deep. Packing arranged in a geometric pattern to provide convoluted flow channels approximately ⅜ in. sq	2 4 6	155 155 155	32.5 65 97.5
Crimped Metal Mesh	Multiple layers of herringbone-crimped brass wire mesh woven from 0.01 in. diam wire and having 16 openings per in.	4	120	51
Straight Glass Fibers	4-8 in. long glass fibers, 135 micron diam, random-packed in planes parallel to cell face; 3.2 lb fibers per cu ft cell volume	2	210	52
Crimped Nylon Fibers	4 in. long crimped nylon fibers, 250 micron diam, random-packed in planes parallel to cell face; 1.17 lb fibers per cu ft cell volume	1.75	81	15
6-Denier Dynel Fibers	Bonded fiber pack containing 0.5 lb of 25 micron diam fibers per cu ft of cell volume (63% fibers and 27% bonding agent by wt)	2	266	55.5
24-Denier Dynel Fibers	Bonded fiber pack containing 0.6 lb of 48 micron diam fibers per cu ft of cell volume (63% fibers and 27% bonding agent by wt)	2	168	35
Curly Glass Fibers	1-2 in. long curled pyrex glass fibers 37 micron diam; random packed 0.67 lb per cu ft of cell volume	2	168	35

closing the instrument. Calibration of the aspiration psychrometer with a sling psychrometer confirmed that readings by the two instruments were identical.

An aspiration psychrometer of identical design was used downstream of the test cell, but this instrument was carefully heat insulated, since otherwise, the air withdrawn from the washer cabinet would have been subjected to warming from the surroundings through the walls of the instrument. According to theory, the wet bulb temperature of the air does not change during passage through the humidification chamber. Therefore, it could be assumed that if heat entered or left the system through the cabinet walls or by way of the water pump, the leaving wet bulb temperature would rise to give warning that the process deviated significantly from adiabatic conditions. In practice, the downstream psychrometer location, shown in Fig. 1, was generally satisfactory; the difficulties experienced at this point were principally connected with wetting of the dry bulb by droplet carryover.

Temperature measurement of the spray water was accomplished by immersing a thermometer in the liquid reservoir below the wet cell. Often, temperature of the sump liquid exceeded inlet and outlet wet bulb temperature by as much as 1 F, indicating heat absorption from the surroundings during passage through copper water tubing plus heating from the mechanical work of the pump impeller. Magnitude of this temperature difference, when it occurred, was not considered a serious deviation from adiabatic conditions within the scrubber.

Facilities were not available for regulating the temperature of makeup water, so water was added on a batch basis and the unit operated until the sump thermometer indicated the bulk liquid had reached the wet bulb temperature of the air before making psychrometric readings. The liquid sump was generally capable of holding sufficient water to permit at least a one hr data run, so that lack of facilities to regulate the temperature of the makeup water was not a serious inconvenience; especially since the 10-15 minute equilibra-

Table II—Spray Nozzles

Nozzle Designation* (in.)	Full or Hollow Cone Pattern	Delivery and Total Spray Angle at 15 psi gpm	degrees
⅜ × 0.209	Full	2.25	87
¼ × 3/32	Hollow	0.29	78
¼ × 1/16	Hollow	0.185	58

* First number refers to thread size for attachment to water piping system; second number refers to nozzle outlet orifice diameter.

tion run, after liquid additions, could be used for making air flow resistance measurements for which water temperature was unimportant.

The wet cells used in this study are identified and described in Table I; the spray nozzles, in Table II.

As a general rule, with centrifugal, hydraulic nozzles, the smaller the nozzle size the smaller and more uniform drops that are formed; and drop size decreases with increasing pressure. Therefore, it may be assumed (and this is in good agreement with observation) that the ⅜ x .209 nozzle produces the coarsest spray drops and the ¼ x 1/16 nozzle the finest; with the ¼ x 3/32 nozzle producing drops of an intermediate size, but more nearly like those of the ¼ x 1/16 nozzle.

Air flow resistance of each cell was measured dry and at one or more water flow rates over a range of velocities from approximately 100 to 600 fpm. Pitot-type static pressure probes (made in conformity with directions contained in the ASME publication PTC 19.2.2,3. 1945 and calibrated with a standard Pitot-static tube in a moving air stream) were installed immediately up-and-downstream of the test cell, and the differential pressure read with an alcohol-filled vertical or 1/10 inclined U-tube manometer. Considerable difficulty was experienced, due to plugging the static probes during wet runs. Coating the exterior of the probes with a film of water-repellent silicone oil caused the moisture to gather together into large .drops which frequently dripped off and, in any case, made them much less likely to bridge over the tiny static pressure holes. Even with this aid, it was generally necessary to blow out the tubes before each reading.

Serial humidity measurements were usually made by varying air flow velocity while holding all other conditions constant. In a few instances air flow was held constant and water rate changed systematically. Air flow resistance measurements were always made by the former method. Percent humidification efficiency "E" was calculated from up-and-downstream psychrometric readings as follows:

$$E = \left[1 - \frac{t_2 - t_{w2}}{t_1 - t_{w1}} \right] \times 100 \quad (6)$$

Although t_{w2} should equal t_{w1} in a

FIG. 2 – HUMIDIFICATION EFFICIENCY VS AIR FLOW

USING 2-1/4" x 3/32" HOLLOW CONE NOZZLES
WATER RATE = 0.6 GPM/SQ FT

CURVE NO.	CELL	CELL DEPTH-IN.	[illegible] SQ.FT.	[illegible]
1	[illegible]	2	32.5	—
2	[illegible]	4	65	—
3	[illegible]	6	97.5	—
4	[illegible]	4	51	284
5	[illegible]	1.75	15	284
6	[illegible]	2	30	30
7	[illegible]	2	20	46
8	[illegible]	2	35.0	25
9	[illegible]	2	52	125

truly adiabatic process, up-and-downstream wet-bulb temperatures did not always exactly coincide (although the discrepancies were small); therefore, Equation 6 was utilized throughout.

EXPERIMENTAL RESULTS

Humidification efficiency – Fig. 2 shows experimental curves of humidification efficiency vs air velocity for nine cells. Each was wetted by two ¼ by 3/32 nozzles delivering a total of 0.6 gal of water per minute; equivalent to 0.46 gpm/sq ft of cell face area. These curves clearly show that humidification efficiency decreases as air velocity increases and that humidification is highly sensitive to small changes in air rate. This behavior agrees with the relationship predicted by equation 4c. Since the water rate was 0.6 gpm during all these experiments, it will be evident that as the air rate changed, the water rate, in terms of gal of water per 1000 cu ft of air was altered, also. Therefore, when air flow through the humidification cell was reduced, not only was residence time within the wetted packing increased, but, in addition, the amount of water available for humidification per cu ft of air was, proportionately, greater. This aspect of the humidification process will be discussed in another section.

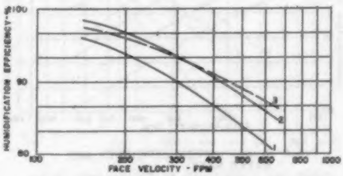


FIG. 3 - HUMIDIFICATION EFFICIENCY OF ETCHED METAL PACKING

USING 2 - 3/8"X.209" SOLID CONE NOZZLES, WATER RATE = 3.6 GPM/SQ FT. CELL FACE

CURVE NO.	CELL	CELL DEPTH-IN.	
1	CRIMPED RIBBON	4	NOT ETCHED
2	CRIMPED RIBBON	4	ETCHED
3	CRIMPED RIBBON	6	NOT ETCHED

It is obvious from inspection of Fig. 2 that certain of the cells were much more effective in humidifying air than were others. It is natural to assume that these differences are inherent in the nature and amount of packing in each of the 9 experimental cells; this information is contained in the Table accompanying Fig. 2. The gross total surface area of the packing material contained in each cell is of especial interest, since theoretical considerations indicate that effective wetted surface is one of the principal variables. Curves 1, 2, and 3 of Fig. 2, representing three different depths of the cell packed with crimped aluminum ribbon, show a steady and significant displacement in the direction of improved humidification efficiency as the gross packing surface increased from 32.5 to 65 to 97.5 sq. ft. We see, for example, at an air velocity of 200 fpm, that unsaturation (i.e. 1-E) decreased from 19% to 14% and then to 10% as the surface of the cell packing increased by equal increments.

The relationship between gross packing surface and humidification efficiency for cells of diverse packing materials, however, is less orderly. For example, curve 4, representing a 4-in. deep cell containing 0.01-in. diam wire mesh and curve 5, representing a 1¾-in. deep experimental cell packed with 0.01

in. diam crimped Nylon filaments, respectively, are similar; yet the Nylon-containing cell has only 2/7 the surface area of the wire mesh-containing cell. Further, humidification curves, 7, 8, 9, though practically identical, represent cells containing from 35 to 55.5 ft of gross packing surface. A consideration of filament diam shows no better correlation with humidification efficiency than does total gross surface area. Clearly, neither total gross surface area nor filament diam may be utilized as a reliable index of humidification efficiency, since the influence of packing arrangement and other factors on the production of effective wetted surface are known inadequately.

The observation, that a given quantity of 254 micron Nylon fibers produced the same humidification effect as 3½ times this amount of metal screening composed of wire of identical diam, suggests that there may have been significant differences in the manner in which the fibers were dispersed. In this case, however, the metal-filled cell was machine packed and the filaments were uniformly distributed throughout the cell space (probably much more so than the handpacked Nylon fiber cell), so that poor distribution of the metal filaments was definitely not a contributory factor to this cell's relatively poor humidification performance. An alternate explanation is thought to be connected with the greater wettability of the Nylon filaments. All the metal cells were oil-coated when received (and properly so, since these units are normally employed as dry dust filters for air conditioning systems). Although they were thoroughly purged prior to making humidification measurements, it is entirely possible that an oil film remained on the surfaces of the packing materials.

It was noted, even after the crimped metal ribbon cell had been well cleaned, that water on the packing surfaces tended to gather into large drops rather than to spread out and wet all the surfaces. This behavior appeared to be characteristic of the smooth, shiny aluminum surfaces of the packing even when the metal was free of oil. It seemed reasonable to believe that humidification efficiency of the cell would be improved if the packing surfaces could be more completely wetted. Therefore, a standard 4-in. deep crimped metal ribbon cell was solvent-degreased and electrochemically etched in order to give a deep

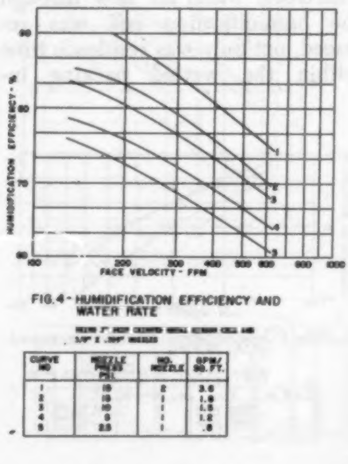


FIG. 4 - HUMIDIFICATION EFFICIENCY AND WATER RATE

[illegible]

CURVE NO.	NOZZLE PRESS PSI	NO. NOZZLE	GPM/ SQ. FT.
1	18	2	3.6
2	18	1	1.8
3	10	1	1.5
4	5	1	1.2
5	2.5	1	.8

satin finish to all metal surfaces and then retested. The before- and after-etching results, shown in Fig. 3, very clearly indicate that etching the shiny surfaces of the metal packing significantly increased the humidification efficiency of this cell; to the degree that a 4-in. deep etched cell became as effective as a 6-in. deep unetched cell. This confirmed the important role of "wetability of the packing" in humidification efficiency.

All the measurements shown in Fig. 2 were made while using two ¼ x 3/32 spray nozzles at 15 psi nozzle-pressure. However, by alternating nozzle pressure, it was possible to make changes in the water delivery rate, and by utilizing different nozzles, it was possible to produce different spray droplet sizes and spray patterns (as well as different water rates).

The effect of water rate and nozzle characteristics on humidification efficiency are illustrated in

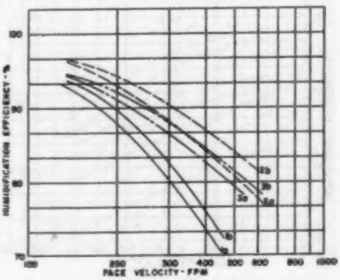


FIG. 5 - HUMIDIFICATION EFFICIENCY AND SPRAY NOZZLE CHARACTERISTICS

USING 2" DEEP CURLY GLASS

CURVE NO.	NOZZLE	NO. NOZZLES	WATER GPM/SQ.FT
1a	1/4" x 1/16"	2	.3
1b	"	"	.4
2a	1/4" x 3/32"	2	.5
2b	"	"	.6
3a	3/8" x .209"	1	1.5
3b	"	"	1.8

Figs. 4, 5 and 6. Fig. 4 shows humidification curves for a 2-in. deep crimped metal ribbon cell when wetted with 1 and two ⅜ x .209 full cone nozzles at spray pressures from 2.5 to 15 psi.

Although spray pattern and average droplet size are likely to change with changes in nozzle pressure, the range of pressure variations employed was restricted sufficiently to suggest that the principal change was in the quantity of water emitted. Humidification efficiency, as shown in Fig. 4, appears to increase in a regular manner as the water rate increased and, in fact, Fig. 4 shows that the percentage unsaturation decreased roughly in proportion to the increase in water rate. For example, at air velocities of 200 and 300 fpm, unsaturation decreased 7 percentage points when water rate nearly doubled from 0.8 to 1.5 gpm; and it decreased 5-6 percentage points when the water rate doubled from 1.8 to 3.6 gpm. This is in good agreement with the effect predicted by equation 4c, if it is assumed that increases in water rate produce a proportional increase in the effective wetted surface area.

In Fig. 5 there are humidification curves for a 2-in. deep cell packed with 35 micron diam curly glass wool and wetted with three different nozzle systems. In Fig. 6 there are similar curves for a 2 in. deep cell filled with 135 micron diam random packed straight glass fibers wetted with two types of nozzles. It may be seen in both Fig. 5 and 6 that, just as was noted above, for a given nozzle system,

humidification efficiency increases as the nozzle pressure (i.e. quantity of water) increases.

However, when the effectiveness of the different nozzles is compared, it may be seen from Fig. 5 that the ¼ x 1/16 nozzles, which deliver the smallest water quantity, were least effective; but that the ¼ x 3/32 nozzles, which deliver only 1/3 the water quantity of a single ⅜ x .209 nozzle, were most effective in producing wetted surface in the cell packing. The absolute superiority of the ¼ x 3/32 nozzle over the ⅜ x .209 nozzle (as well as on a gal-for-gal basis) is also shown in Fig. 6.

It may be concluded from the evidence in Figs. 5 and 6 that water quantity is only one of the important factors for judging nozzle performance and that the size of the spray droplets which are formed is of equal, or greater, significance. Extreme fineness of droplet size is not necessarily beneficial, since fine droplets are likely to pass through the cell packing without impacting and producing wet surfaces. On the other hand, when water is broken up into large droplets only, there is an insufficient number to adequately wet all areas of the packing.

Humidification curves for 2, 4 and 6-in. deep crimped metal ribbon cells of identical construction were seen in Fig. 2; it was pointed out that the humidification efficiency improved as the gross surface area of the packing (i.e. cell depth) increased. Similar results were observed when these same cells were wetted with two ⅜ x .209 nozzles (instead of two ¼ x 3/32 nozzles as shown in Fig. 2), and curves of humidification efficiency vs cell depth for selected air flow rates are shown in Fig. 7. These curves are in close agreement with the relationship between humidification efficiency and cell depth (z) indicated by theory, as shown in Equation 4c.

It was noted during the discussion of the humidification curves shown in Fig. 2 that when air flow changed, the water-air ratio also changed (because the spray system maintained a constant minute volume water rate). Other experimental data indicated that when air flow was held constant and the water rate varied, important changes in humidification efficiency also occurred. To separate the effect of these two interrelated factors, all the pertinent humidification efficiency data were recalculated in terms of water rate (in

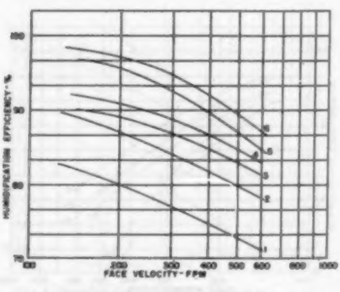


FIG. 6 - HUMIDIFICATION EFFICIENCY AND SPRAY NOZZLE CHARACTERISTICS

* USING STRAIGHT GLASS FIBER CELL

CURVE NO.	NOZZLE	NO. NOZZLES	WATER GPM/SQ.FT.
1	3/8" x .209"	1	.8
2	"	1	1.2
3	"	1	1.5
4	"	1	1.8
5	1/4" x 3/32"	2	.3
6	"	2	.6

gals) per 1,000 cu ft of air flow and replotted in Fig. 8. Data developed with different numbers of nozzles and over a full range of nozzle pressures have been consolidated in the humidification curves in Fig. 8, and the fact that both droplet size and spray pattern were somewhat affected by these changes has been ignored.

In spite of these variations, plots of water-air ratio vs humidification efficiency gave smooth curves which indicated that over the range of air velocities studied (i.e. 150-600 fpm), humidification efficiency was independent of air flow and proportional to water-air ratio, alone. For each cell, it appears that number of nozzles and spray water pressure affect humidification efficiency only insofar as they influence the quantity of water delivered to the cell.

Since the relatively large range and diversity of experimental data represented by the curves of Fig. 8 (as well as the close similarity of curves for each combination of nozzles and cell analyzed in this fashion) indicate that most, if not all, cell packings exhibit similar behavior, it is desirable to inquire more closely into the manner in which the amount of water affects humidification.

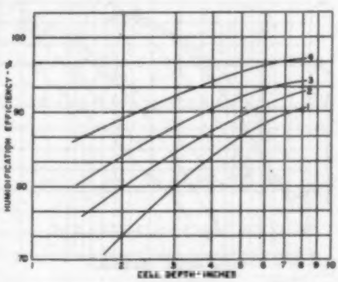


FIG. 7 - HUMIDIFICATION EFFICIENCY AND CELL DEPTH

[illegible]
WATER RATE = 3.6 GPM/SQ FT

CURVE NO.	VELOCITY - FPM
1	200
2	300
3	400
4	600

First of all, humidification efficiency is closely tied to the amount of wet surface in the cell, i.e. effective area of packing. It is reasonable to believe that increased water rates produce more effective wetting of the available packing surface. Experiments with baffled water cooling towers and with gas absorption towers packed with rings and saddles bear this out, and it has been concluded that the fraction of the packing surface effectively wetted depends mainly on the liquid rate; only very slightly on the gas velocity.

Over the much narrower range of air velocities studied in these humidification experiments, the influence of velocity undoubtedly becomes exceedingly small. Further, increased water rate may affect humidification efficiency indirectly by narrowing the air passages and thereby increasing the local velocity in the interstices of the fiber packing. Examination of factors affecting the magnitude of water vapor and heat transfer coefficients indicates that increased air velocity acts beneficially in this respect.

Therefore, it must be recognized that small changes in water rate (all other conditions remaining the same) result in complex changes in the total effective wetted area

of the packing, as well as in the magnitude of the transfer coefficients. Theoretical analysis of the humidification mechanism indicates that humidification efficiency is inversely proportional to some function of the air velocity (equation 4c) but, in practice, the magnitude of the opposite change in effective wetted area and transfer coefficients may more than offset the velocity effect.

Referring to the curves in Fig. 8 for the 2 in. deep curly glass cell packed to a density of 0.67 lb per cu ft, it may be seen that for equal water rates the ¼ x 1/16 nozzle was more effective than the ¼ x 3/32 nozzle, and both were more effective, gal for gal, than the ⅜ x .209 nozzle. During the discussion of Fig. 5, it was noted that two ¼ x 1/16 nozzles were less effective than two ⅜ x .209 nozzles. However, this is true only when comparing performance on a nozzle-for-nozzle basis; since it is clear from Fig. 8 that gal-for-gal the smaller nozzle is far superior.

It is a well-noted fact that small nozzle orifices produce finer spray droplets and, consequently, a greater area of air-to-water interfacial surface per gal of water sprayed. This holds true for the period during which the droplets are air borne and, probably (though this is by no means certain), for the period during which they reside on the packing surfaces themselves. From this, it may be concluded that spraying the same water volume from ¼ x 3/32 or ¼ x 1/16 nozzles would effect a significant and substantial increase in humidification efficiency of the wet cell over that produced by the ⅜ x .209 nozzle. For example, it may be seen from Fig. 8 that substituting ¼ x 3/32 for ⅜ x .209 nozzles, increased humidification efficiency of the curly glass filled cell from 82 to 92% at a water rate of 2.5 gpm/1000 cu ft; while use of ¼ x 1/16 nozzles raised humidification efficiency to 95% under the same operating conditions.

The use of fine droplet spray nozzles increases the tendency of free water to penetrate the humidification cell and it is likely that the use of fine spray nozzles will be most successful with cell packings of small diam fibers which possess good stopping characteristics for small diam water droplets.

Air flow resistance—Cell resistance is also an important factor in practical humidifier applications. When cells have the same air flow resistance, there is obvious merit in

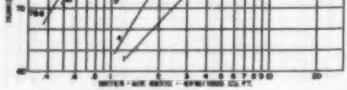


FIG. 8 - HUMIDIFICATION EFFICIENCY AND WATER - AIR RATIO

CURVE NO.	CELL	NOZZLES
1	[illegible]	3/8" x .209"
2	[illegible]	"
3	[illegible]	"
4	[illegible]	"
5	[illegible]	"
6	[illegible]	1/4" x 3/32"
7	[illegible]	"
8	[illegible]	1/4" x 1/16"
9	[illegible]	1/4" x 3/32"
10	[illegible]	"

selecting the one with greatest humidification efficiency (assuming cost, maintenance and other factors are roughly equal).

For example, the glass straight fiber cell had the same air flow resistance as the Nylon cell, but higher humidification efficiency (e.g. 93% vs 86.5% at 380 fpm), and would, therefore, appear to be a better choice for cell packing, unless the Nylon fibers should prove to possess unusually good self-cleaning ability or some other desirable property.

Similarly, a choice could be made between a 4-in. deep crimped metal mesh and a 4-in. deep crimped metal ribbon cell, solely on the basis of cost, since humidification efficiency and air flow resistance were closely equivalent.

Experience with fibrous filters suggests that air flow resistance of washer cells should be directly proportional to cell depth and an examination of air flow resistance curves in Fig. 9 for three thicknesses of the crimped metal ribbon cell wetted with two ¼ x 3/32 nozzles at 15 psi water pressure bears this out.

The air flow resistance curve for each of the cells appearing in Fig. 2 is shown in Fig. 9. It is particularly significant that the relative order of the curves is quite different in these two Figs. (indicating that certain of the cells are capable of producing air humidification with lower power requirements).

There is a temptation to combine, in some manner, data on air flow resistance and humidification efficiency and to derive a "figure of merit" value for each cell, or cell-nozzle combination, which rates units on an overall basis such as "percentage points of humidification efficiency per hp of energy expended on air and spray water."

While figures of this sort are always of considerable interest, they fail to properly account for the relative importance of the two factors in any given situation. In

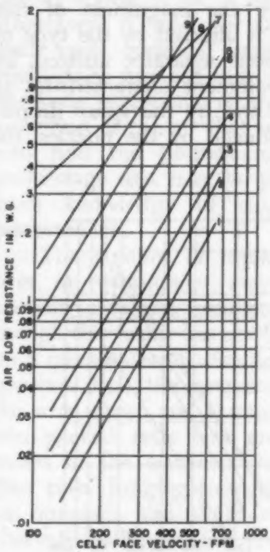


FIG. 9 - RESISTANCE AND AIR FLOW

USING 2 - 1/4" X 3/32" HOLLOW CONE NOZZLES. WATER RATE = 0.6 GPM/SQ.FT.

CURVE NO.	CELL	CELL DEPTH IN.	PACK'G DIAM. MICRONS
1	CRIMPED RIBBON	2	—
2	CRIMPED MESH	4	254
3	CRIMPED RIBBON	4	—
4	CRIMPED RIBBON	6	—
5	STRAIGHT GLASS	2	135
6	CRIMPED NYLON	1.75	254
7	24-DEN. STEEL	2	48
8	CURLY GLASS	2	35
9	4-DEN. STEEL	2	25

certain instances the "merit figure" would have to be heavily weighted in favor of air flow resistance (e.g. application of humidification apparatus to existing systems) while in others, the reverse would be true (e.g. industrial constant humidity chambers). Under these circumstances it is thought preferable to consider each factor separately.

The quantity of water sprayed onto the face of the cell has an important influence on air flow resistance, just as it has for humidification. In Fig. 10 are pressure drop-velocity curves for a 1.75-in. deep Nylon fiber cell when dry and when wetted at 4 different water rates with two ⅜ x .209 nozzles. It may be noted from the Fig. that the cell had greater resistance when wetted than when dry and that each incremental increase in water rate produced a smaller and smaller increase in air flow resistance. This same pattern was noted with other cells, as well. It is believed that most of the increased energy loss represents energy transfers from air to liquid during the process of blowing water droplets off the cell surfaces since the large air passages of cells like the crimped metal ribbon are not likely to be greatly restricted during wetting. The percentage increase in air flow resistance due to cell wetting appears to be roughly the same over the entire range of velocities tested, and the slope of the resistance-velocity curves was approximately 0.6; close to fully-turbulent flow, which would have had a slope of 0.5.

DISCUSSION OF RESULTS, SUMMARY AND CONCLUSIONS

All the tests which have been described refer to a single wet cell without an entrainment eliminator. The use of an entrainment eliminator of one type or another is common practice in commercial wet cell washers. As its function is to remove liquid droplets, it is reasonable to expect that the surfaces of the entrainment eliminator will become wetted to some extent and that increases in air humidification will occur.

Since the magnitude of this increase is affected by the type of entrainment separator utilized, by the entrainment characteristics of the cell and by the spray droplet size produced by the nozzles, the

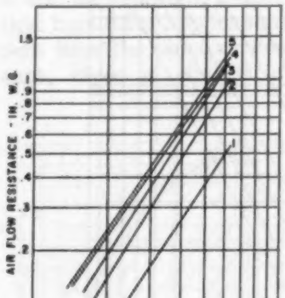


FIG. 10 AIR FLOW RESISTANCE DRY AND AT VARIOUS WATER RATES

USING 1.75" DEEP CRIMPED NYLON CELL

CURVE NO.	GPM/SQ. FT.
1	0
2	0.6
3	2.4
4	3.0
5	3.6

use of eliminators during these tests, designed to evaluate the basic efficiency of a single humidification cell, would have been a needless complication; and so they were omitted.

Identical wet cells are often. used in series for increased effect. Theoretical considerations suggest that the magnitude of the heat and mass transfer coefficients, discussed previously, does not change as the humidification process progresses toward saturation; and test results appear to be in agreement. Therefore, it may be concluded that each identical stage of a multi-stage wet cell unit produces the same percentage change in the condition of the air that reaches it and that the performance of a multi-stage unit may be predicted from knowledge of single cell performance.

The amount of liquid carryover, or entrainment, may be important in certain installations. Although this factor has not been put on a quantitative basis, there is no question that the carryover from the wire screen and aluminum ribbon packed cells was greatly in excess of the carryover from the fiber cells. Judging from the large air openings and slight offset of alternate baffle sections present in these cells, large amounts of liquid carryover are not surprising. In practice, this type of cell should always be followed by a moisture eliminator, preferably a fine fiber dry pad, when absence of free moisture in the humidifier outlet is critical.

Perhaps the most interesting information to come out of this investigation is recognition of the negligible effect of air flow rate on humidification efficiency over the range of velocities investigated. Humidification efficiency was found to be directly related to the ratio of water to air and to the water surface area produced by the spray system.

This suggests a number of interesting possibilities: first, that considerable improvement in humidification efficiency can be achieved by maintaining the same spray volume but changing from coarse spray nozzles to fine spray nozzles. The magnitude of the humidification increase achievable by this means appears to be of far greater value than the modest increase in cost for the extra piping and nozzles. Second, that considerable increases in washer air flow capacity appear to be practical when increased spray rates are utilized.

The principal deterrent to substantially greater air flow rates is the corresponding increase in air flow resistance. However, there seems to be great interest in high pressure heating-ventilating systems for office buildings, hotels, apartments, etc., in which high pressure, high velocity air streams are conveyed to the point of utilization and there expanded and mixed prior to release to the occupied space.

It seems unlikely that increased air flow resistance of a washer would be a serious obstacle in systems of this type and the distinct advantages of smaller, cheaper and

larger capacity units for the same humidification efficiency are obvious. Doubtless these factors are of considerable significance in conventional systems as well. Certainly, for commercial and industrial installations, increased air flow resistance should be a minor hindrance compared to the advantages enumerated above.

For applications where humidity conditions close to saturation at exit from the washer are not required, the use of cells such as the crimped metal units may be advantageous because of quite low air flow resistance and ease of installation. Where virtually complete humidification is required, two stages of crimped metal cells followed by an entrainment separator would produce a humidification efficiency of 98.5%, while two stages of fiber cells would produce a humidification efficiency of 99.5%.

Whether or not this difference in humidification efficiency is commercially significant would be an important consideration in the selection of one cell or another. The above figures refer to an air flow rate of 400 fpm; other air flow rates would produce different, but comparable, results. In cases where equipment cost is an important factor, it may be possible to obtain the desired outlet conditions with a single stage of fiber cells, whereas two stages of crimped metal cells would be needed. For this type of requirement, the somewhat higher air flow resistance of a straight glass cell would more nearly equal the resistance of two crimped metal cells in series.

Air flow resistance curves for the various cells and nozzle systems indicated that cells, when wet, have appreciably greater resistance than cells when dry, and that increasing the water rate increases cell air flow resistance. Resistance also rises with increasing air rate, greater packing density of the fibers, depth of filter, and decrease in fiber diam.

DISCUSSION

T. Kriho, Denver, Colo.: Has any research been incorporated using a wetting agent in the water with the different media?

Author First: No experiments were carried out on that, but I believe the effect would be similar to that experienced after etching the aluminum cell. That is to say, if a cell material was difficult to wet, an increase in efficiency would result if, by adding a wetting agent to the water, the water spread more effectively over the packing material. If the packing material was already well wetted, the effect would be negligible.

K. R. Goddard, Savannah, Ga.: Has any experimental work been done with cascading the cells to increase efficiency and perhaps lower wet bulb temperature?

Author First: No, these experiments were done with single cells to keep the system as simple as possible. If two cells are placed in series, the same percentage effect will be obtained in the second cell as in the first. This is true in other gas absorption equipment and for other mass transfer processes of the same nature.

Walter Cole, Minneapolis, Minn.: Is there a relationship between efficiency on the filter and velocities tested? Is there any area in which a maximum of efficiency was achieved?

Author First: The data indicated that for a constant water rate, efficiency decreased as the air velocity increased. The efficiency-velocity curves did not show an area of maximum efficiency for air velocities ranging from 150 fpm to approximately 600 fpm.

ARTHUR GROSSMAN, Denver, Colo.: An eliminator was not shown; were accurate wet bulb readings obtained?

AUTHOR FIRST: The principal problem was to prevent wetting of the dry bulb thermometer by liquid carryover from the wet cell. A representative fraction of the air stream was drawn into the downstream aspiration psychrometer through a tube placed at 90 deg to the main flow. This prevented entry of water droplets. Nevertheless, it was necessary to check the dry bulb frequently and to dry it, when necessary.

F. H. BRIDGERS, Albuquerque, N. M.: Was the effect of nozzles without pads tested?

AUTHOR FIRST: No, but there probably would be a considerable humidification effect with nozzles alone. The nozzles were placed fairly closely to the cells so that the time the droplets were air-borne was brief and the major effect was in the cell itself.

F. E. INCE, St. Louis, Mo.: Has the water rate vs. the gaseous rate been plotted at a relatively good efficiency, say 80%, to determine how this may be balanced economically? We are interested in the economic values of the size of the equipment, together with the operating cost, including space and equipment costs. These questions may be resolved by plotting them at relative efficiency value.

AUTHOR FIRST: Some information relating to this subject was considered qualitatively and it was concluded that smaller equipment and smaller nozzles, but more of them, at higher pressures could probably be used. However, when going to higher air and water pressures, problems of noise and vibration may develop.

ROBERT ASH, Phoenix, Ariz.: For given dry and wet bulb temperatures, what effect does a change in total pressure have on the humidifying efficiencies for constant mass velocity of air and for constant linear velocity of the air?

AUTHOR FIRST: All the experiments reported upon were done at atmospheric pressure so there is no experimental evidence to offer. However, theory indicates that vapor diffusivity is inversely proportional to absolute pressure. Diffusivity, in turn, influences the transfer coefficient; but other factors, such as air and liquid rates, have greater influence for moderate changes in pressure.

1736

No. 1736

Metastable State of Water in Relation to Heat Exchangers

BURGESS H. JENNINGS

Investigations were carried out in an effort to determine whether conditions associated with the metastable state might produce explosive actions of sufficient intensity to cause rupture of boiler walls or piping equipment. Evidence obtained rather strongly indicated that serious damage from this was highly improbable, but under conceivable circumstances, not impossible. After a careful search of the literature, a number of experiments were carried out. This investigation which was started at the request of the Technical Advisory Committee on Hot Water and Steam Heating* took place at Northwestern University as a cooperative research investigation sponsored by ASHRAE.

GENERAL PATTERN OF METASTABILITY

Tables of the properties of fluids have been prepared for almost all of the substances in common use, such as water-steam, refrigerants, many hydrocarbons and other organic liquids. Such properties always indicate the temperature-pressure (t-p) relationships which indicate equilibrium saturation conditions. For example, with water-steam (H_2O), appropriate tables of the properties of steam (Ref. 1) would show that at 14.696 psi pressure, a temperature of 212 F must be reached before boiling takes place. The term boiling, as used here, means the orderly phase change associated with a liquid transforming into vapor (or the re-

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H. C. Day	H. A. Lockhart
W. S. Harris	M. W. McRae
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E. G. Keller	M. H. Westerberg

verse of vapor condensing into liquid). Similarly, at 100 psi, the boiling temperature is 327.8 F; at 200 psi, 381.8 F; and at 400 psi, 444.6 F. The question, why these particular values might be raised and is it always true that for water-steam the values are invariant? Although these values are those of equilibrium and statistically probable, it is possible for aberrations from these normal values to exist. Moreover, when a fluid has deviated from its equilibrium properties in relation to phase change, it is not completely stable, and it has become the custom to speak of the fluid as existing in a state of metastability.

It has long been recognized that, although equilibrium conditions, as indicated above, are the most usual and relatively stable conditions, it is possible for water to exist in other configurations. This is true not only with the phase change relating to evaporation but also with the phase change which occurs when a liquid solidifies.

For example, it is not difficult to chill water carefully to temperatures below 32 F without the formation of ice. Normally, when heat is abstracted from water which has reached 32 F in temperature, ice starts to form and further heat extraction results in the formation of more and more ice, with the temperature remaining at 32 F, until all of the liquid has turned into solid after which, if further heat is abstracted, the temperature of the solid then falls, with cooling occurring in the solid phase.

Sometimes, however, if the heat abstraction takes place rather slowly and quietly, the liquid, on cooling to 32 F does not change phase as further heat is abstracted, but readily drops in temperature to values as low as 24 F; even reaching 20 F. If, however, this liquid, subcooled below 32 F, is disturbed by stirring or other shock, the metastable state is upset, ice crystals form throughout the mixture, and the temperature immediately rises to 32 F, with normal freezing again taking place. Even when no forced disturbance takes place in the subcooled water, usually at about 8 to 12 deg below 32 F the unstable state rectifies itself, the mixture forms ice crystals and moves back to equilibrium or to what might be considered the most stable form.

This condition of metastability, in relation to the phase change from liquid to vapor, occurs in somewhat similar fashion because, if gas-free water in a smooth-surface vessel is quietly heated, it is possible to raise the temperature of the water far above the normal equilibrium pressure and temperature. That is, in an open vessel, water (at 14.7 psi) reaching the 212 F temperature will normally boil, but occasionally it may stop boiling and rise in temperature 2 or 3 deg above 212 F, after which stabilization will occur; and temperature drops back to 212 F.

If care to add heat at a uniform and slow rate is exercised, the same pattern applies, but temperature increases of 12 to 15 F above equilibrium occur before readjustment. This phenomenon can

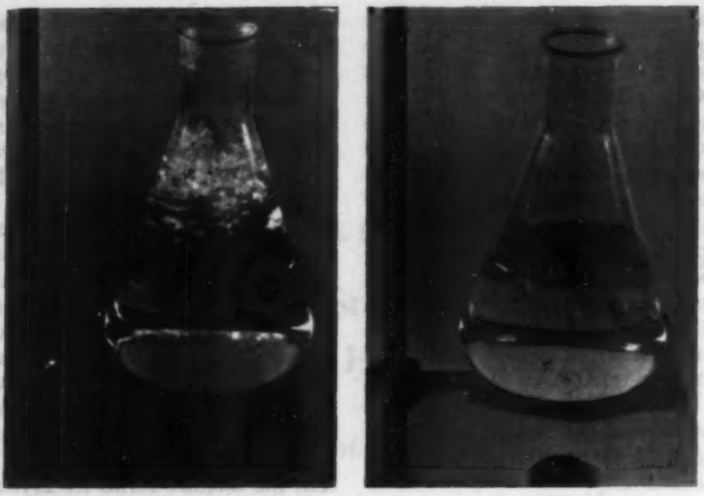


Fig. 1 Water boiling in flask under normal conditions (left). Metastable boiling of gas-free water (right)

be observed with ordinary glass or metal vessels, provided they have smooth walls. Using water in a capillary tube, Kenrick, et al, (Ref. 3) were able to heat water to a metastable temperature of 518 F at atmospheric pressure. The work of Kenrick took place under conditions that would not be applicable to engineering operations.

The process of bubble formation in a liquid is a complex problem which is closely related to surface tension of the liquid, as well as the other factors of phase change. Surface tension naturally resists the bubble formation and is thus important in the boiling process. The equation expressing the relationship between the pressure inside of a bubble and the surrounding liquid is developed in most thermodynamics texts[3] and in treatises on physics. In simple form it appears thus:

$$\Delta p = \frac{2s}{r}$$

where Δp represents the pressure difference between that inside of a bubble and that outside of the bubble in contiguous superheated liquid; S is the surface tension of the liquid; and r is the radius of the bubble. With quite small bubbles where r is only a micron or so, calculation will show that the pressure differential can approach values reaching to hundreds of psi. Surface tension decreases with increasing temperature of the fluid and reaches a value of zero at the critical temperature.

These comments are not in-

tended to create the impression that boiling a liquid is a difficult problem, because it is not the case. Nevertheless, if it were not for a number of compensating features, the boiling process would be much less satisfactory in equipment than we find it to be. As has been mentioned, boiling is helped by dissolved gases, and by the convection currents, which in connection with surface roughness can alter flow to the point of making boiling less difficult. During the boiling process some superheat (metastability) always exists in the liquid.

A corollary to boiling is the problem of effervescence in liquids which are saturated or supersaturated with gas. This condition exists in carbonated beverages, where it can easily be observed that if the bottle containing the beverage is vigorously shaken, a rapid and intense release of the gas from solution takes place. This is similar to what happens when bubbles form in a superheated liquid, for, in this case also, the generation of bubbles is accelerated whenever the body of the liquid is disturbed.

EXPERIMENTATION IN GLASS VESSELS

The first phase of the investigation involved laboratory studies designed to see how readily the metastable state could be produced. A series of experiments was started with the work being done completely in glass. As has been mentioned, in ordinary boiling the process is greatly helped by gases dissolved in the water, and it is also aided by surface roughness of the vessel which contains the water.

Chemists, for example, often put glass beads in the bottom of a beaker undergoing heating, for this practice promotes smoother boiling. A conclusion from this would be that both the rough surface of the beads and bubble nuclei on the surface of the beads act to promote bubble formation and smoother boiling. It is true almost without question that bubble nuclei adhere to the surface of such beads, existing in the form of adsorbed gas. There is, however, strong doubt that beads or other objects, because of surface effects or rounded points, contribute to bubble formation. In fact, it is possible to observe sharp points, broken glass lying in the bottom of a beaker, and to note that there is more evidence of bubbles being formed on the smoother surfaces of the glass than at the points.

If such broken glass or beads are completely boiled out to the point of being essentially degassed, they apparently are not particularly effective as bubble inducers. It is in another way that the beads serve in the formation of bubbles, because superheated (metastable) water has difficulty maintaining its superheated or metastable condition if the water is agitated as by continual stirring or rapid motion. In an assemblage of beads in the bottom of a beaker, it is true that thermally-actuated convection currents are necessarily active, and the scrubbing or convective action of the currents over the surface of the beads acts to cause bubble formation from the resulting turbulence.

In carrying out early testing, distilled water was put into Erlenmeyer flasks and boiled for a long enough period to remove the greater portion of the dissolved gases in the water. The only difficulty encountered was to prevent all the water boiling away before the gases had been removed. After some trials, it was possible to accomplish this, and an almost completely degassed water was obtained. When the rate of heating the water was then greatly reduced, boiling in normal fashion would stop and the condition of metastable boiling which is also known as boiling with bumping began.

Under boiling with bumping, the water no longer boils at equilibrium temperature but tends to rise a few degrees above equilibrium. Following the temperature rise, a readjustment takes place and vapor forms with sufficient violence to jar the container and create an audible sound, which explains the name attributed to this type of boiling.

Considering normal atmospheric boiling temperature of 212 F, it is possible, in an open flask of this type, using a Bunsen burner, to reach metastable temperatures as high as 218-220 F. This appeared to be the top limit which could be reached in a simple trial of this type using a Bunsen flame in an open room.

Fig. 2 Flask of water boiling under metastable conditions, using side radiant heating

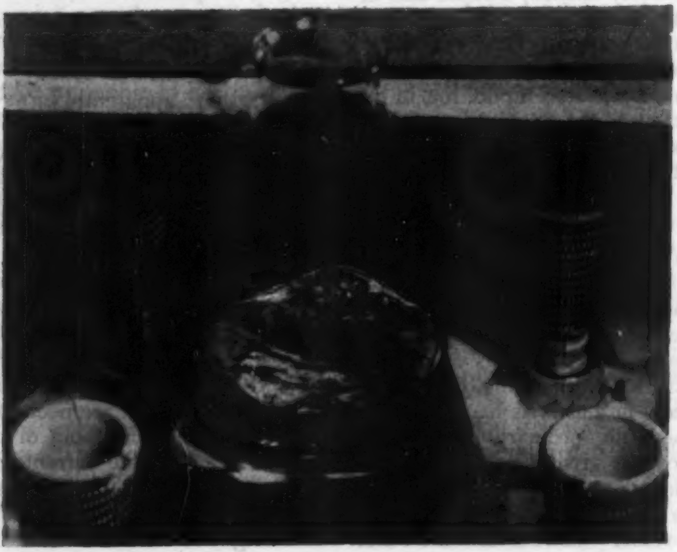


Fig. 1 is a picture of two Erlenmeyer flasks operating under similar conditions. At the left, water is shown being heated and boiled in normal fashion. This involves ebullition in the true sense of the word; that is, a number of bubbles move throughout the mixture, steam is released at the upper surface and passes out to the top of the flask. At the right, metastable boiling is taking place. This involves a surging action in the liquid, and the liquid appears to swirl around. Almost no bubbles are present, and steam leaves the water surface in limited amounts.

Following the tests with distilled water, a number of similar tests were made using ordinary city tap water, as there was doubt whether tap water with 100 to 200 parts per million of solids in solution, and possibly more gas also in solution, would behave in a fashion similar to distilled water. The investigations showed that city tap water behaved in almost the same manner as distilled water and, in a superficial way, showed the same limits of metastable temperature rise.

The third step in this program with glass was to set up a more carefully controlled environment for heating in order to find out what limit of metastable temperature rise might be reached. For this purpose, a circular well-type structure some 6 in. high was built, into which the flask could be placed, and inside this space a number of radiant electric heaters were inserted. The character of the enclosure and the heaters can be seen in Fig. 2, with a flask in position.

For this part of the investigation, a flask, partially filled with degassed water and at boiling temperature, was moved into the container and subjected to continued heating from the radiant electric heaters. Using these, the controlled heat input largely entered from the sides of the flask. Without much difficulty temperatures as high as 228 F could be reached under free atmospheric conditions. This represents a 16 F metastable temperature rise over normal boiling temperature. At these higher temperatures, it became increasingly evident that the return to equilibrium, when it occurred, was usually violent.

Moreover, some event was usually needed to trigger the return to equilibrium; and having learned this, a better basis of test control was established. For example, if metastability were present, when the water was stirred by a thermometer to take a temperature reading, the metastable state would be immediately upset. It was found that such stirring was inadvisable, as a scalding jet of water shot from the flask as the return to equilibrium took place. Consequently, two other methods were used to trigger the return to equilibrium. The first was to stir the water with a hooked wire when the upset was desired, and the second was to drop a small pellet into the flask to cause the same result.

Fig. 3 shows what can happen when this return to equilibrium occurs. This is a synchronized

Fig. 3 Triggered explosions of metastable water from a flask surrounded by radiant heaters

photograph, which followed the dropping of a pellet in the flask when the water temperature was at approximately 227 F. The water in the flask re-established equilibrium with great violence, forming steam and throwing water and steam upward about 5 ft, sufficiently high to wet the ceiling of a 9-ft room. Although it was difficult to photograph this result clearly, the streaks indicating the water and steam being expressed to the ceiling are clearly visible in the photograph.

TRIALS IN PRESSURE VESSELS

The next phase was to investigate metastability in closed metal vessels at greater than atmospheric pressures. Here the problem of degassing the water and of having the metastable state exist with water contacting rough-surface metal walls existed. A small bomb-like device was constructed for this. See Fig. 4. It consists of a length of 6-in. pipe capped at one end and provided with a flanged connection on the other end. Into the flange head a diaphragm device was prepared to record pressure surges and a mercury manometer was connected. A Bunsen burner heated the liquid contents through the container walls of the vessel, a bare thermometer also dipped into the liquid contents, and arrangements for filling and emptying the tank with appropriate valves and tubing were also provided.

In operation, the tank was essentially filled with previously degassed water which was heated

and further degassed by venting off air and steam from the container. A small amount of make-up water was added to replace the amount which had been boiled off and the process repeated several times until it was felt reasonably certain that the liquid in the vessel was essentially gas-free. At this point a metastable investigation could be started, and when water almost completely filled the chamber, the liquid was heated to a temperature of approximately 240 F; heating was then stopped. The temperature of the water and the pressure indicated by the mercury manometer were both carefully noted.

Cooling of the uninsulated chamber immediately set in, and the question at hand was whether a metastable condition existed in the chamber. Corresponding readings of pressure and temperature showed deviations from equilibrium which were not particularly different, and cooling and pressure drop occurred at a slow rate. It was noted that the attached mercury column instead of dropping uniformly would sometimes undergo an oscillation and then continue its uniform descent. While researchers wondered what caused this oscillation, the pipe was inadvertently jostled: it was noted that an abrupt and sharp oscillation in the mercury column took place.

Further investigation showed that during the cooling process if the vessel were sharply struck by an object, such as a block of wood or hammer, an oscillation in the

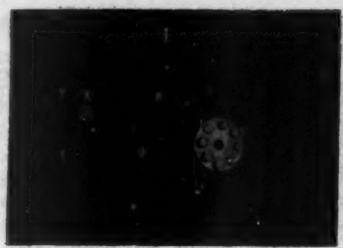


Fig. 4 Pressure vessel employed in metastable water tests

mercury column nearly always followed. However, once the oscillation had occurred, striking the vessel a second time produced no effect; that is, apparently a metastable state existed in the liquid during the cooling process which, disturbed by the shock, brought the system immediately to equilibrium, and equilibrium having been established, further striking of the vessel showed no pressure surge effect.

This phenomenon was easily reproducible and all that was required was gas-free water and a cooling process of the type described. The amount of water in the tank was found not to be critical, but a vapor space was always provided. Actually, the fuller the tank the more violent the shock effect, the extent of which, measured by the mercury manometer, was only 2 to 5 in. of mercury in magnitude. Shock effect (short-time pressure peak) from a metastable state to equilibrium is of greater magnitude than this, but for the low temperature differences involved, the resultant pressure

surge will probably always be small.

Consequently, the pressure diaphragm which had been placed on the flange was not usable, as its stiffness ratio had been set for significant deflection in a range for pressure differences of 25 psi or more. Note that either distilled or tap water could be used interchangeably in the pressure vessel and that the same type of results were obtained. Thus, there was no evidence that material in solution in the water prevents the condition of metastability.

In summary, this vessel had shown data of the following type:

Warming at 240.0 F at 51.0 in. Hg (essential equilibrium)

Cooling in 236.0 F at 46.1 in Hg (1.2 in. unbalance indicated)

Striking the vessel in the range of 236 F showed distinct oscillations of the mercury columns reaching about 2.5 in. per leg from the undisturbed state

Striking the vessel immediately after such a readjustment showed trivial oscillations, about 0.3 in. per leg max

Oscillations in reducing magnitude (i.e. less than 2.5 in. per leg) as the temperatures fell toward 212 F and the vessel was struck

Mild oscillations in the subatmospheric region continued to some 190 F (about 0.2 in. per leg)

Manometer oscillation from straight mechanical shock with cold water in the vessel were hardly noticeable and the movement was 0.1 in. per leg or less

The results of this work were sufficiently encouraging for the existence of the metasable state, to indicate that further work should be carried out. The vessel described, because of its limited liquid volume, the high heat capacity of its walls and the necessity of redesigning its temperature-pressure measuring system if precise indications were to be obtained, was redesigned for continuing the investigation.

Because a somewhat practical device was desired, an ordinary 30-gal galvanized-iron, hot-water domestic service boiler, provided with pressure taps, filling and venting devices, an electric heater, and with essentially the same items that were provided for the original pressure bomb, was used. Previously degassed water filled the tank, and there was great care to further degas the water in order to produce the proper environment for metastability.

The unfortunate fact is that metastability could not be found to exist. This conclusion was reached only after three months of intermittent testing, in which every conceivable means was tried in an effort to show the existence of metastability. Whether the relatively rough inside surface of the galvanized metal resisted degassing so effectively that metastability could not occur, or whether some other condition prevented its appearance, the fact remains that it was not possible to show any evidence of the phenomenon. Consequently, with some regret, the test on this tank was discontinued and another design was planned and made.

Rough surface was suspected to be a significant factor, so the next step involved a structure built from a smooth-walled material. For this purpose a vertically mounted piece of 4 in. ID brass pipe was selected. This was 36 in. long, and the ends were closed by brazing a ¼-in. brass plate to each end.

Heating was produced by a 25-watt electric heater screwed into the tank through a tapped opening in the bottom plate. The heating rate was controlled by varying the heater voltage with a variac. The tank was equipped with a thermometer well, sight glass, manometer pressure connection, and with a ½-in. pipe outlet at the top, which ran to a surge or air-venting vessel. Connection to the air-vent outlet was by a rubber hose fixed to a water-cooled glass condenser. The outlet of the condenser connected to a ¼-in. copper tube about 20-ft long, the far end of which dipped into the glass bottle that served as a surge tank.

In operation, the tank was filled with degassed water, and this water was then boiled by the electric heater in the tube vessel. Steam and gases from the water passed up through the condenser where nearly all of the steam condensed, and any steam that was not condensed passed through the copper tube finally to condense in the water of the surge tank.

The prime requirement was to provide water as gas-free as possible, and the arrangement described forced the steam rising from the water surface to reverse its direction until it passed through the condenser. It was felt that air entrained in the water would largely be driven off by continuous boiling. Passing through the condenser, the steam was partially condensed, the condensate ran back by gravity, while the remaining air and steam was forced into the surge tank to bubble through the water in it.

After a long period of boiling, to insure that essentially all of the gases in solution had been removed, all valves on the tank were closed; and the contained water was heated until the pressure reached 50 in. of mercury. This value was used because it corresponded to the max range of the attached manometer. When the pressure was reached, the heater was turned off and the pressure lowered in the tank several ways.

The first method consisted of opening a small needle valve connected to the tank, to allow steam to escape to reduce the pressure gradually to atmospheric. Next the tank transferred heat to the surrounding air and thereby cooled. Another was to cool the tank with forced convection generated by an electric fan blowing on the tank. The final method was suddenly to pour water over the outside of the tank. This produced quick condensation and pressure unbalance beyond metastable effects.

Each method caused a condition which was interpreted as metastable, evident from the sudden surge of mercury in the manometer, which occurred when the wa-

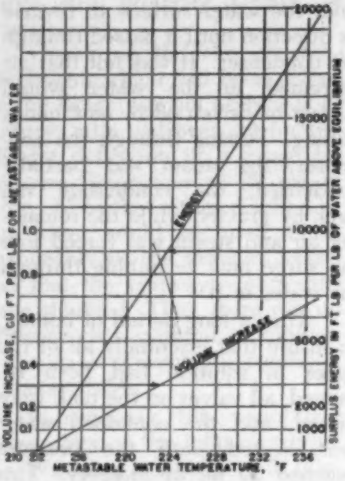


Fig. 5 Surplus energy and excess volume as related to equilibrium for metastable water at atmospheric pressure (14.7 psi)

ter reverted to equilibrium and produced a sudden rise in pressure as the unbalance corrected itself. It was also found that, in the case of the tank slowly cooling, if the outside were tapped, the metastable state would be disturbed and a surge of pressure, amounting to a total of 8 in. of mercury, appeared in the manometer.

When the tank was allowed to cool, without being struck externally, the manometer would surge periodically indicating that the cooling process was not a continuous line of equilibrium states but rather an alternation of metastable and equilibrium states as the cooling proceeded. The pressure rises observed under normal cooling without upset were approximately half as great as those which occurred when the tank was struck.

Runs were made with the water level in the tank at different positions, from almost full to 50 per cent vapor space by volume. Magnitude and intensity of the surges were not altered in relation to the amount of water in the tank. Magnitudes of the surges are thought to be higher than indicated on the mercury manometer, but it is doubtful that they are greatly in excess of those indicated. Considering the smallness of the metastability surge, it was not felt necessary to develop a low-pressure, precision-diaphragm pick-up for the pressure measurement.

METASTABLE CALCULATIONS

A study of energy conditions in metastable water can be readily made with the help of the steam tables, and the results of such a study are shown in Fig. 5. This figure is set up for atmospheric pressure and on the assumption that the specific heat of superheated water is not seriously different from saturated water in the same temperature range. The surplus energy of superheated water over that of saturated water at the same pressure has been plotted. Thus, at 212 F, equilibrium exists, and no surplus energy is indicated. If, on the other hand, the water when heated to 220 F remains as a liquid, it will have picked up 8.0 Btu/lb or approximately 6300 ft-lb of excess energy. Similar values appear on the chart for higher temperatures.

When metastable water at a 200 F temperature is triggered, and it flashes to water and steam at equilibrium temperature (212 F), the surplus energy serves two functions: First, to change the phase or provide the internal latent heat for the vaporizing portion of the water; second, to perform the external work required as the vaporized water reaches its equilibrium volume.

For example, at 220 F, water at atmospheric pressure in flashing would increase in volume by 0.022 cu ft/lb, assuming the volume increase to take place at atmospheric pressure. The external work, as the medium expands, amounts to 47 ft-lb/lb. This value in one sense is a measure of the work produced in readjustment from the metastable condition. However, it does not truly measure the work function, and for this purpose it is better to make use of the so-called availability function:

$$\Delta B = (E_2 + p_0 v_2 - T_0 S_2) - (E_1 + p_0 v_1 - T_0 S_1)$$

which is applicable to a process of this type. Its use is explained in advanced thermodynamics books such as Ref. 5 and Ref. 6; and if this function is evaluated for the same conditions, it is found that the max useful work, derivable from this change of state, amounts to some 39 ft-lb/lb. Although this numerical illustration was worked out for atmospheric pressure conditions, similar patterns would apply for higher pressure levels.

In terms of energy relations for water and steam, these work values are relatively small and lead one to believe that the resultant energy releases under metastable return to equilibrium are also small, so that the pressure rise in a restricted system would not be of great magnitude. However, if exceptionally high metastable temperatures did occur, the conditions would be quite different, as the reestablishment of equilibrium could induce a severe pressure rise or surge followed by readjustment to a final equilibrium pressure higher than the initial metastable pressure.

Consequently, it is not difficult to imagine conditions under which a steam boiler or a water heater could operate for a sufficiently long period that the resultant water could be almost completely degassed; and if the same degassed condition were true for the boiler surfaces, true metastability might arise. If the metastable temperature spread were large, a significant shock could occur. However, the experimental realization of such conditions is so difficult as to indicate that a metastable explosion in severe form is highly improbable.

CONCLUSIONS

Two things are evident from these tests. First, in normal cooling, the metastable temperature was never many degrees away from that corresponding to pressure equilibrium; second, the magnitude of the shock is relatively small, probably in every case less than 5 psi.

The reason for this investigation was to determine whether me-

tastability created a danger or hazard in the operation of equipment. For the following reasons it does not:

1. Most actual installations contain so much gas in solution in the water that the requirements for serious metastability are not present.

2. With quite rough surfaces, we were unable to show the presence of metastability.

3. The magnitudes of the pressure changes that we experienced were in general below those that would be expected to lead to failure.

However, on the other side of the question, it must be stated that pressure pulses can be caused by metastability; and it is possible to imagine a set of circumstances under which such pressure pulses could be severe enough to cause damage.

The author has been unable so far to find an authenticated case of rupture of a steam (hot-water) boiler produced by metastability, and this is also true for pressure vessels containing other fluids. However, reports that certain systems do show strong symptoms of metastability have come to our attention.

In one instance, under certain conditions, during the start-up period an evaporator which presumably contained only liquid ammonia, oil and ammonia vapor frequently was subjected to what appeared to be a violent shudder. At this time the evaporator was vibrating, and the pipes connected to it were violently disturbed.

Similar phenomena have been reported for some of the high-vacuum, water-salt refrigeration systems on start-up after a period of non-operation. In the systems described, long periods of operation occurred during which any entrained gas would most likely have been vented or purged from the system, and tube surfaces had the opportunity of becoming relatively smooth or coated.

Time can be a factor in the metastability phenomenon, especially when a liquid close to its saturated condition is suddenly allowed to expand and the consequent formation of vapor is necessary. A number of experimental investigators (Refs. 7, 8, and 11) have conclusively confirmed that the amount of vapor which may be formed is not the amount that would be predicted by equilibrium thermodynamic theory, from which it can be deduced that a liquid core in the flow stream remains in metastable condition while the remainder of the stream undergoes the conventional partial vaporization we have come to expect.

ACKNOWLEDGMENTS

This investigation of metastable water was carried out over a considerable period of time with my former close associate, Professor Willard L. Rogers, taking a leading part in much of the original work. In addition to his help, a number of graduate assistants were extremely active and their help is also acknowledged.

Particular thanks are given to

Armando Torloni, Milan Jovanovich, and William T. Snyder for their experimental work and analysis. The Institute of Boiler and Radiator Manufacturers was most generous in support of this research. Members of its committees worked closely with the Technical Advisory Committee on Hot Water and Steam Heating and recommended many of the experimental patterns which were followed.

REFERENCES

1. J. H. Keenan and F. G. Keyes, Thermodynamic Properties of Steam, John Wiley and Sons, New York, 1936.
2. M. W. Zemansky, Heat and Thermodynamics, McGraw-Hill Book Co., First Edition 1937, p. 257-260.
3. F. B. Kenrick, C. S. Gilbert, and K. L. Wismer, Jour. Phys. Chem, Vol 28, p. 1297, 1924.
4. R. B. Dean, The Formation of Bubbles, Jour. of Appl. Physics, Vol. 15, May, 1944, p. 446-451.
5. J. H. Keenan, Thermodynamics, John Wiley and Sons, New York, 1941.
6. B. F. Dodge, Chemical Engineering Thermodynamics, McGraw-Hill Book Co., New York, 1944, p. 74-76.
7. R. S. Silver, Temperature and Pressure Phenomena in the Flow of Saturated Liquids, Proc. Royal Soc. of London, Vol. 194, series A, 1948, p. 464-480.
8. P. F. Pasqua, Metastable Flow of Freon-12, Refrig. Eng., Vol. 61, October, 1953, p. 1084A-1088.
7. R. S. Silver, Temperature and Pressure Bubbles in Superheated Liquid, Jour. Appl. Phys., Vol. 25, April, 1954, p. 474-8.
10. M. S. Plesset, S. A. Zwick, Growth of Vapor Bubbles in Superheated Liquids, Jour Appl. Phys., Vol. 25, April, 1954, p. 493-500.
11. W. T. Bottomley, Flow of Boiling Water Through Orifices and Pipes, Trans. Northeast Coast Institute Engineers and Shipbuilders, Vol. 53, 1936-7, p. 65.
12. S. G. Bankoff, R. D. Mikesell, Growth of Bubbles in a Liquid of Initially Nonuniform Temperature, Trans. ASME Paper 58-A-105, 1959.

DISCUSSION

W. BROWN, Lexington, Ky.: In one paper dealing with a specific acid, the authors were working with liquid ether; they first boiled the ether for some time, not only to degas the liquid but to remove the vapor. They then cooled the liquid below the boiling temperature and increased the rate of energy supply (or raised the temperature again). In this manner it was possible to achieve a super heat of about 200 F in the liquid, until it reached a point where a violent explosion occurred. The diameter of the tube is not known, but it probably was a small beaker. Another condition was that the surface was extremely smooth initially.

AUTHOR JENNINGS: That is metastability, unquestionably. The diameter of the tube is an important factor here.

R. G. RANEY, Columbus, Ohio: On the basis of experiments carried out, it was found that absolutely pure liquids can be superheated considerably; in some tests a liquid superheated from 80-100 F. However, if a noncondensable gas was dissolved in the liquid, then it would boil at the correct temperature. The capillary tubes used had an inside diameter of 15/1000 in.

Tests were also run in glass tubing with a diameter of 1/10 in. On this type of application it was found that if there is a noncondensable gas in the liquid and it is kept there, it is possible to boil at the correct temperature and pressure.

T. LEITERSDORF, Hamilton, Canada: Has any relationship been observed between the rate of heat, the rate of firing or rate of generating heat and the source of heat relative to the volume of the container which was used? It would seem likely that there would be some difference in observation between one state and another.

AUTHOR JENNINGS: Convection currents will dissipate metastability. No matter what character the vessel has at the working surface, if there is motion where the heat is coming in and the active localized superheat occurs, metastability will be broken up. Therefore, in order to get metastability, the rate at which that heat comes into the surface should be lowered so that the convection activity is broken up to a point where no metastability occurs. This is true in connection with either large or small containers.

T. LEITERSDORF: There are two conditions that have been observed for some time which may fall within the realm of metastability. One is that of priming and boiling, where there is a sudden eruption of water and steam with the specific condition of design; perhaps from a rapid rate of firing. The other is the peculiarity of the sudden removal of the refrigerant gas with oil. This has been quite evident in connection with the new small, compact compressors. Here there appears to be a direct relationship between the volume of the container being heated or cooled and the size of openings in pipes that remove the liquid.

AUTHOR JENNINGS: Priming and this refrigeration aspect are not necessarily metastability. When there is a large mass of liquid at high temperature and the pressure is suddenly lowered, for whatever reason, the surplus energy in this liquid must go somewhere. This, in a sense, is the metastability pattern but it is an externally controlled condition. A boiler explosion should not be referred to as an instance of metastability; there is an enormous mass of liquid and if for some reason a tube fails, the pressure is reduced and the surplus energy in the liquid has to appear somewhere. However, it is not metastability that caused the explosion but a controlled external reaction.

C. M. ASHLEY, Syracuse, N. Y.: A number of tests has been run at low temperatures primarily with water, both in freezing and evaporation, and even under dynamic conditions metastability would be present almost invariably, provided that there was no air in the material. If there is an airfree mixture of water under freezing conditions, usually between 1 and 3 deg subcooling results. In the case of boiling at low temperatures, it is usual that no flash occurs and no visible boiling occurs even at a 5 to 10 deg temperature difference. What happens in those conditions is that there is a certain effect due to the fact the liquid is in droplet form; also, there is undoubtedly a considerable amount of subcooling at the surface which also contributes to it. However, when air is present, that situation no longer holds. Metastability of refrigerants is well known; the refrigerant laws are calculated on the basis of liquid flow.

AUTHOR JENNINGS: Has this problem also been encountered in connection with salt solutions, to your knowledge?

C. M. ASHLEY: Yes, in scaling-water solutions, where the water is boiled. For example, this applies to sea water and relates to some extent to the presence of impurities, although the impurities do help to cut down on the metastability. It was found necessary to have the solid phase present to be sure of avoiding metastability.

1737

No. 1737

Flow and Heat Exchange Characteristics of Finned Tube Exchangers

B. GEBHART

Heat transfer and friction loss characteristics of a surface are determined by the exact nature of the flow process over the surface. There are two basically different kinds of flow possible near the surface of a solid, separated and unseparated flow. In unseparated flow fluid moves parallel to the surface and high velocities are found quite close to the surface. In separated flow the region of well ordered flow is removed from the surface, and intervening space is filled with a fluid of varying velocity moving about in a complicated way.

Unseparated flow occurs, for example, on a flat plate placed in a stream at zero angle of attack. Both separated and unseparated flow are found on blunt bodies, such as a cylinder or plate normal to the flow.

Various measurements of local heat transfer on surfaces having regions of separation have indicated that heat transfer rates are not vastly different for separated and unseparated portions of the surface.[1] Heat transfer rates for a long cylinder and for a plate at zero angle of attack are essentially equal for ordinary fluid velocities, if heat transfer areas are the same per ft of span, if $\pi D = 2L$.

The principal effect of separation is to cause flow losses. Separation produces a region of low pressure on the back side of the surface and results in a high form drag. Total drag, which is an indication of flow losses, is the sum of frictional and form drag. For a cylinder and typical flow conditions in air, the form drag is over 90% of the total. For a zero angle of attack plate there is no form drag; all flow losses are due to frictional forces. Therefore, cylinder flow losses are of the order of ten times as great as those of the plate with the same area per ft of span.

B. Gebhart is Associate Professor of Mechanical Engineering, Cornell University. This paper has been prepared for presentation at the ASHRAE Semiannual Meeting, Chicago, Ill., February 13-16, 1961.

Comparisons of relative heat transfer and flow losses demonstrate that separation is an undesirable effect, if flow losses are important. The point of view taken in this work is that high heat transfer rates and low flow losses are both important.

EFFECTIVENESS RATIO

Effects of fluid separation indicate a need for a quantity which combines heat transfer and flow loss characteristics of a surface into a single parameter. A number of methods have been introduced in the past to permit such comparisons. See, for example, the discussion in Ref. (2). However, since these parameters do not combine the heat transfer and friction loss characteristics in a simple, direct way, a quantity called "effectiveness ratio" was defined.

The effectiveness ratio is heat transfer effectiveness divided by flow loss effectiveness. Heat transfer effectiveness, ε_E, is the amount of heat transferred divided by the max amount that an ideal exchanger would transfer, given the fluid temperatures.

$$\varepsilon_H = \frac{U_o A \Delta t_a}{m c_p \Delta t_m}$$

The flow loss effectiveness, ε_E, is defined as the flow energy dissipation rate divided by the kinetic energy of the fluid streams. In this study of air flow over finned tubes, ε_E is based upon air side losses alone.

$$\varepsilon_E = \frac{\Delta p_f}{\dfrac{\rho_a V^2}{2g_c}}$$

The effectivenesses are dimensionless, as is their ratio N, the effectiveness ratio.

$$N = \frac{\varepsilon_H}{\varepsilon_E}$$

The effectiveness ratio depends upon the heat exchanger geometry and, in general, is a function of the fluid flow Reynolds number and the fluid Prandtl number.

It appears to be a good indicator of the desirability of a geometric arrangement. Streamlined surfaces generally have small separated regions and, therefore, high values of N. Blunt or bluff bodies have large separation regions and small values of N, as shown in Table I. These values are based upon actual heat transfer and pressure drop information obtained from standard references. The flat plate at zero angle of attack is a

TABLE I. EFFECTIVENESS RATIO FOR THE FLOW OF AIR OVER ISOTHERMAL SURFACES OF VARIOUS GEOMETRIES

Surface	N	Flow Condition
Flat Plate at zero angle of incidence	0.63	Laminar Boundary Layer
	0.63	Turbulent " "
Turbulent Flow inside a tube	0.70	All Reynolds numbers
Flow normal to a cylinder	0.141	$N_{Re} = \dfrac{U_o D}{\nu} = 100$
	0.072	$= 1{,}000$
	0.018	$= 10{,}000$
	0.009	$= 100{,}000$

streamlined form and causes no separation at all. Turbulent flow inside a tube has no separation. Values for a circular cylinder are much lower, due to separation.

Similar calculations, based on the performance of several types of deformed and relieved fin coils, show that the effectiveness ratio is also a sensitive indication of the flow pattern over this type of exchange surface. A flat fin has a relatively low N, undoubtedly because a portion of the fin surface contributes only friction losses. Slots in the fin do not cause important separation and a high value of N is found. Rippling of the fin would be expected to cause separation and results in a low value of N.

VISUALIZATION OF FLOW PATTERNS OVER FINNED TUBE EXCHANGERS

Since the nature of the flow pattern over a heat transfer surface is of crucial importance in determining relative heat transfer and flow loss characteristics, a study of the flow pattern was first carried out. Models were studied in a visualization tunnel by using smoke filaments. This technique is especially suited for determining flow separation.

Models of finned tube coils were made: (Fig. 1). These studies were designed to show whether flow separation on the fin causes deformed fins of the rippled type to be poorer than fins relieved by slotting. Models of the good (slotted) and the poor (rippled) deformed fins were built and studied for evidence of separation and boundary layer relief. The upper two fins in the model shown were flat Lucite; the fin under study was put in the third position.

Visual observations with smoke filaments indicated that the flat and slotted fins produce no separation and that the rippled fin has extensive separation behind every sharp corner. The various smoke observations may be summarized as follows:

(1) Back half of the tubes were always separated flow.
(2) Fin surface in the wake of the tube is an area of high turbulence.
(3) Flow over all other flat fin surfaces is smooth and unseparated.
(4) Separation occurs over all sharp corners.

CONSIDERATIONS IN DESIGNING RELIEVED FINS

The preceding sections indicate observations and arguments in which the point of view adopted was that a good deformed fin design is one which maximizes the effectiveness ratio. Although a large value of N indicates specifically high heat transfer with low pressure loss, it may also imply certain other things as well, for example, that unseparated flow produces little noise. Separation and consequent vortex and turbulence production result in greater noise.

The builder of the exchange surface is vitally concerned with many aspects of extended surface which have nothing directly to do with effectiveness ratio (for example, low material cost, simple con-

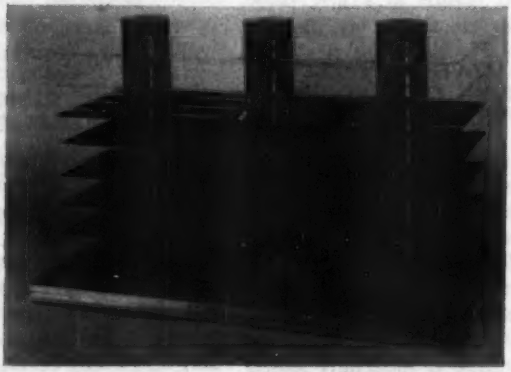
struction, and low maintenance). However, since decisions on the relative importance of such factors are necessary to obtain a truly optimum practical design in each particular circumstance, and because the relative weights of the various factors change from one case to another, a simpler point of view was adopted in this study. A high effectiveness ratio was sought using fin designs which require no more material than the plain flat fin and which are as simple to assemble. Better fin shapes seemed possible and practical within these limitations.

RELIEVED FIN DESIGNS AND TESTS

Many particular fin types were considered. The fins tested were designed to provide compact, low flow loss, high heat transfer rate surface. Separation was to be avoided.

Coils having six different relieved fin designs were built, tested, and compared with the flat fin coil in the air velocity range from 200 to 1000 fpm. Tests were carried out in the draw-through tunnel diagrammed in Fig. 2. The coil specifications and fin types are shown in Fig. 11. The tube and fin

(a)



(b)

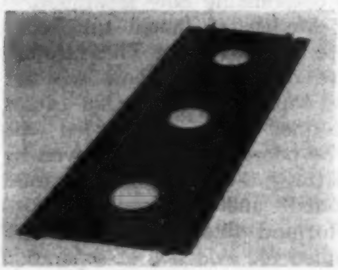


Fig. 1 Model fins and finned coil
(a) Coil with slotted fin in test position
(b) Rippled fin
Tube diam 1.30 in. Fin width 3-1/3 in.
Tube spacing 3-1/3 in.
Fin thickness 1/64 in.

materials and methods of manufacture and assembly were the same for all coils.

Coil A is the flat fin against which comparisons were made. Coil B is the first relieved fin design and is based upon punching out reliefs in a flat fin. The fin areas farthest from the tubes were removed. Slots and holes were provided to interrupt the fin boundary layer. The fin requires only 75% as much material and has only 64% as much area as the plain flat fin.

Coils A and B were tested in a low turbulence air stream at air velocities of 200, 400, 600 and 800 fpm in the following six arrangements:

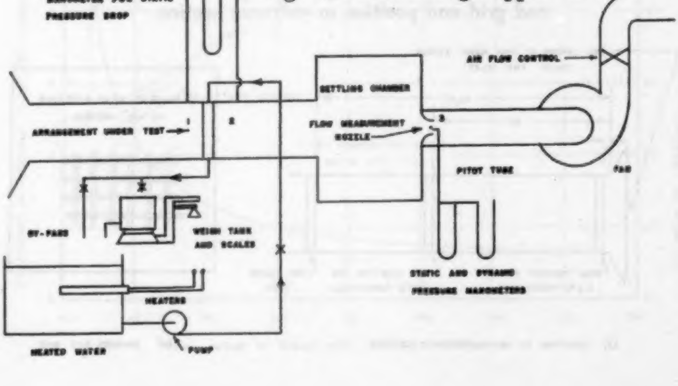
A—Single flat fin coil
B—Single relieved fin coil
C—Two flat fin coils in line, spacing 1 in. between coil centers
D—Two relieved fin coils in line, spacing 1 in. between coil centers
E—Two flat fin coils staggered, spacing 1 in. between coil centers
F—Two relieved fin coils staggered, spacing 1 in. between coil centers

Test Series A and B were a comparison of the two fins in a low turbulence stream. In Series C and D the disturbances produced by the first row tubes mainly encounters the second row tubes and in Series E and F mainly encounters the second row fins.

The nature of the boundary layer relief and the results of the tests of the multi-row arrangements of Coils A and B suggested that the relieved fin would benefit from turbulence in the air stream. Since considerable turbulence is usually present in actual heat exchangers, the single relieved fin coil was tested at air velocities of 400, 600, 800 and 1000 fpm for two turbulence intensity levels.

The draw-through tunnel was modified by placing several layers of fine mesh screen over the inlet and by locating a turbulence producer in the upstream section as indicated in Fig. 3a. The turbu-

Fig. 2 Draw-through apparatus



lence producer was a crossed rod grid made of ¼-in. wood dowelling on 1¼-in. centers, Fig. 3b.

The turbulence level produced by the grid at the location of the coil under test varies with the distance between the grid and coil. If M is the rod spacing in the grid and x is the distance between the grid and the coil, turbulence intensity at the coil is approximately as shown on Fig. 4. This is the decay curve for a grid having a diam-spacing ratio (i.e., D/M) of 0.2 as established by Dryden[3] and later investigators. There is some uncertainty in the use of these results in a different set-up, but the agreement of various investigators of turbulence decay supports the procedure. Compare with the unpublished results of Davis discussed by Sato and Sage.[4] Turbulence intensity, i, is defined as the root-mean-sq average of the turbulent disturbances expressed as a percentage of the bulk velocity of the stream.

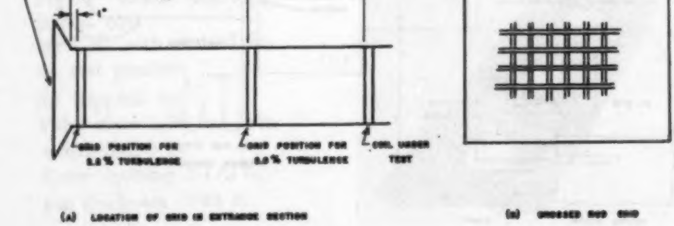
The flat fin was not re-run; it is expected that the turbulence levels employed would have little effect on its performance.* Two series of tests were carried out for a single relieved fin coil. The first series, labelled B′, had a turbulence intensity of 5.0%; the second series, labelled B″, had an intensity of 2.2%.

Results of the tests outlined in the preceding paragraphs guided the design of five additional relieved fins. Each fin was designed to test a particular effect.

The five designs, designated G, H, I. J, and K are pictured in Fig. 11. Fin G is merely fin B with the trailing edge relief and forward holes omitted. Fin H is fin G with the back holes omitted. Fin H was designed to test the postulate that, due to boundary layer build-up, the area between the tubes on the trail-

* This may be inferred from the work reported in Ref. 4 and 5. For example, in Ref. 4 intensities of the order of 5.0% produced a change in the heat transfer coefficient of only 8% on a sphere.

Fig. 3 Turbulence producing crossed rod grid and position in entrance section



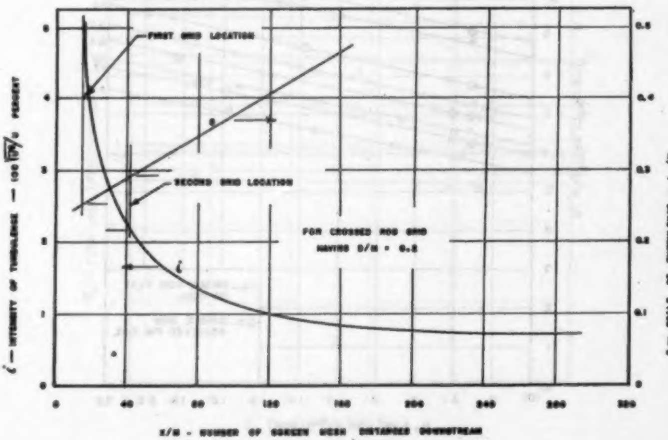
ing edge of the fin would have relatively low fluid friction and be an area of relatively high temperature. The value of the forward holes was doubted, but the back holes were left in fin G to check the effect of boundary layer interruption in the cylinder wake.

The remaining three fin designs (I, J, and K) were different surface relief patterns applied to a fin with a slightly less pronounced leading edge relief. It was thought that the leading edge relief for B (and for G and H as well) was perhaps excessive and that some effective heat transfer area had been removed. Compare leading edge dimensions for coil B and I in Fig. 11.

Coil I has the most boundary layer relief. A new boundary layer begins on the raised portion and on the fin surface behind the relief. In Coil J there is a single relief. The fin design for Coil K was an attempt to reduce flow losses by reducing flow separation behind the cylinder. It was hoped that the raised tabs would direct flow in behind the cylinder and that reduction in flow losses would more than offset the reduction in heat transfer which would result from eliminating the heat transfer area.

Each of the five new fin types (G, H, I, J, and K) were tested in a single coil arrangement in the draw-through tunnel at turbulence intensity levels of approximately 2.2% and 5.0%. For 2.2% turbulence, each coil was tested at 200,

Fig. 4 Intensity and scale of turbulence as a function of downstream distance for a crossed rod grid (from Dryden et al. NACA Report 581, 1937)



600 and 1000 fpm air velocity and for 5.0% turbulence, at 600 fpm air velocity.

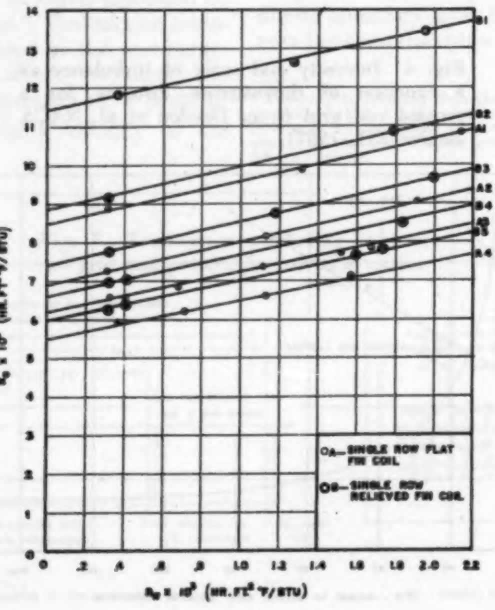
RESULTS

Coils and arrangements indicated in the preceding section were tested under the various air velocity and turbulence conditions stated. For each test condition the coil performance was extrapolated to zero water side heat transfer resistance (i.e., "infinite" water velocity). This permitted a comparison of behavior which did not include the effect of the water side convection coefficient.

The quantity R_o was extrapolated to obtain R'_o. R_o was based upon the same area for all coils, i.e., the inside tube surface area. Calculations are outlined in the Appendix.

For each test several runs were taken at different water velocities. The value of R_o was calculated from the measured heat transfer rate and plotted against water side resistance R_w, calculated from Equation 3 in the Appendix. The resulting curve should be a straight

Fig. 5 Overall resistance vs water side resistance series A & B (for single coil, flat and relieved fins)



line of slope 1.0 since:

$$R_o = R_a + R_m + R_w \text{ and } \frac{dR_o}{dR_w} = 1.0$$

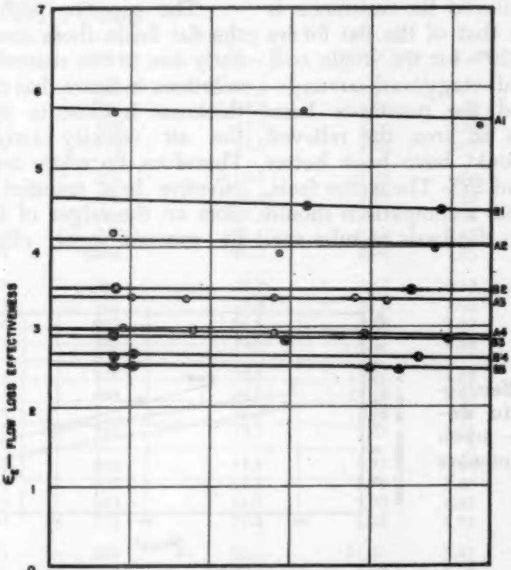
Symbols

The spread of the points from a straight line was extremely small, due, apparently, to excellent heat balances. The slopes of all the best fit curves are within a few deg of the average of their slopes, which was 1.05. See, for example, Fig. 5. All curves were subsequently drawn at this slope. For some of the later tests, measurements were taken for only one water flow rate, the highest which permitted sufficiently accurate measurement of water side temperature change.

The flow loss characteristic for each air velocity and arrangement was calculated by methods outlined in the Appendix. Values for various water velocities at a given air velocity were extrapolated to zero water side resistance. The curves had an approximate zero slope, as expected. See Fig. 6.

Data for the low turbulence test of single coils A and B is plotted and extrapolated in Figs. 5 and 6.

Fig. 6 Flow loss effectiveness vs water side resistance for Tests A & B



Number designations (A1, A2, etc.) indicate the approximate air velocity level. The number is multiplied by 200 to obtain the velocity in fpm. The extrapolated values, R'_o and ε'_B, and the calculated values of ε'_H and N′ are listed in Table II. The value of ε'_H is calculated from the last quantity of Equation 4 of the Appendix.

Results for the six arrangements A, B, C, D, E and F in Table II show many interesting features. For flat fin coils, the two row staggered arrangement has the lowest resistance, followed by the single row. Relieved fins behave in the same way. This difference may be caused by the disturbances of the first row tubes passing over the second row fins.

The relieved fin resistance is higher than that of the flat fin by 30, 30 and 20% for the single coil, in-line, and staggered arrangements. Had the resistance been based upon fin area, the relieved fin coils would have been better by 15, 15 and 25%. The writer feels, however, that a comparison should be made on the basis of tube surface.

The foregoing discussion of the values of R'_o applies also to the heat transfer effectiveness ε'_H, since it is determined directly from R'_o.

Flow loss of the relieved fin is much lower than that of the flat fin in all arrangements. Therefore, relieving has produced an undesirable effect upon resistance and a desirable effect upon flow losses. The effectiveness ratio, which shows the net effect is plotted in Fig. 7 for the single coil arrangements A and B. The decrease in flow loss does not offset the increase in resistance. The flat fin is better in all arrangements, the difference being, 10, 5 and 5% for the single coil, in-line, and staggered arrangements.

The superior performance of the flat fin in these tests is apparently due to two reasons. First, calculations indicate that a .008-in. fin thickness is close to optimum for the air velocity range studied. Therefore, fin edges are still quite effective heat transfer areas, and slots on the edges of the relieved fin remove much effective area.

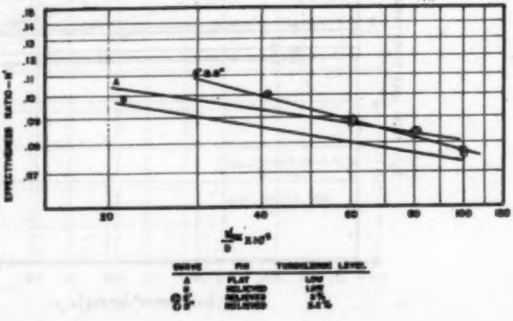


Fig. 7 Effectiveness ratio dependence upon Reynolds number

Second, the tests were carried out in a low turbulence tunnel. Although both fins would benefit from turbulence, it appears certain that the relieved fin would benefit more. Turbulence would disturb the boundary layer as it passes over the slots and holes.

Tests were carried out at turbulence intensity levels of 2.2 and 5.0% for a single, relieved fin coil. These results were extrapolated to zero water side resistance. The results are collected in Table II.

Turbulence reduces flow loss and enhances heat transfer for the relieved fin. Resistance to heat transfer is still higher than that for a flat fin, but the pressure loss is much less. The net effect of these two conflicting tendencies is seen from a comparison of N′ in Fig. 7. Turbulence has substantially improved the performance of the relieved fin coil and has made it somewhat better than that of the flat fin coil at low velocities. This is a substantiation of reasoning which predicted such a result.

However, improvement is not

TABLE IIA EXTRAPOLATED RESULTS FOR SEVEN FLAT AND RELIEVED FIN COIL ARRANGEMENTS OF COILS A AND B FOR LOW TURBULENCE

Test	Approximate inlet air temperature F	V_1 — Air Velocity ft/min	$(N_{Re}/D) \times 10^{-3}$ N_{Re}/ft	$R'_o \times 10^3$ ft²hrF/Btu	ϵ'_{II}	ϵ''_{II}	N′
A1	79	203	19.5	8.55	5.77	.629	.109
A2	77	405	39.0	6.93	4.13	.388	.0938
	72	588	57.5	6.16	3.38	.298	.0881
A4	71	785	77.3	5.48	2.99	.253	.0846
B1	74	203	19.9	11.38	4.58	.467	.1018
B2	76	405	39.4	8.89	3.53	.300	.0850
B3	76	592	57.3	7.47	2.92	.247	.0846
B4	77	790	76.6	6.60	2.69	.208	.0773
B5	79	1000	96.0	5.95	2.55	.184	.0722
C1	77	206	19.9	9.64	10.19	1.089	.1068
C2	77	402	39.8	7.76	7.02	.692	.0986
C3	74	582	56.9	6.85	5.60	.540	.0964
C4	76	786	75.9	6.01	4.83	.460	.0952
D1	71	209	20.6	13.34	7.65	.763	.0998
D2	71	404	39.9	10.18	5.65	.518	.0918
D3	69	583	57.7	8.88	4.50	.415	.0922
D4	69	780	77.1	7.61	3.95	.362	.0917
E1	73	203	19.9	9.27	10.84	1.140	.105
E2	74	400	38.8	6.79	7.88	.795	.101
E3	74	583	56.7	5.79	6.66	.641	.0962
E4	74	773	75.0	5.03	5.92	.557	.0940
F1	70	204	20.1	11.07	9.43	.951	.1008
F2	71	403	39.6	7.98	7.18	.667	.0942
F3	68	578	57.4	6.98	6.00	.527	.0878
F4	71	779	76.4	5.84	5.26	.472	.0895

sufficient to consider this relieved fin better than the flat fin. Given thickness of the fin material, this design is not superior, since considerable fin area has been removed.

The coils G, H, I, J and K were also tested at turbulence intensity levels of 2.2 and 5.0%. The heat transfer resistance and flow loss effectiveness were extrapolated to zero water side resistance; the results tabulated in Table II. The heat transfer effectiveness and friction loss effectiveness for 2.2% turbulence are plotted in Figs. 8 and 9. Also shown are curves for the flat fin (Coil A) at low turbulence and for the first relieved fin (Coil B) at 2.2% turbulence.

Fig. 8 shows that Coil I is superior to Coil A in heat transfer rate for all velocities and that Coil J is superior in the high velocity range. All other coils transfer less heat, per unit tube length, under similar flow conditions.

Fig. 9 shows that only Coil K has higher flow losses than Coil A. That is, all designs other than K have lower flow losses than A. Failure of K is evidently due to separated flow behind the tabs.

Effects of heat transfer and

TABLE IIB. EXTRAPOLATED RESULTS FOR SINGLE COIL TESTS AT 2.2% AND 5.0% TURBULENCE

Test	Approximate inlet air temperature F	Velocity ft/min	$(N_{Re}/D) \times 10^{-3}$ N_{Re}/ft	$R' \times 10^3$ ft²hr°R/Btu	ϵ'_{H}	ϵ'_{II}	N'
B" 3	68	600	60.2	7.5	2.69	.236	.0877
B" 4	68	799	80.2	6.8	2.38	.196	.0824
B" 5	67	1000	98.9	6.25	2.28	.172	.0756
B' 2	6k	411	41.4	9.1	2.75	.276	.1002
B' 3	78	604	58.1	7.4	2.69	.236	.0877
B' 4	75	801	77.8	6.7	2.44	.205	.0840
B' 5	72	996	98.6	6.15	2.28	.175	.0768
G" 1	65	200	20.1	10.46	4.31	.50	.116
G" 3	68	588	59.2	6.83	2.747	.261	.0957
G" 5	64	970	100.2	5.48	2.432	.195	.0802
G' 3	66	585	58.7	6.44	2.70	.281	.104
H" 1	76	212	20.6	9.76	4.603	.519	.1128
H" 3	75	607	58.9	6.55	2.881	.271	.0940
H" 5	76	996	96.4	5.29	2.439	.206	.0844
H' 3	78	585	56.6	6.81	2.721	.271	.0997
I" 1	71	230	22.7	7.98	5.016	.590	.1175
I" 3	75	596	58.1	5.57	3.301	.3255	.0987
I" 5	72	994	97.6	4.62	2.801	.2345	.0838
I' 3	72	595	58.4	5.66	3.175	.3188	.1004
J" 1	72	204	20.1	8.50	5.296	.617	.1165
J" 3	70	593	58.3	5.82	3.251	.3110	.0957
J" 5	71	973	95.4	4.78	2.780	.2310	.0832
J' 3	73	594	58.2	5.81	3.239	.313	.0966
K" 1	76	212	20.6	8.84	6.162	.573	.0931
K" 3	75	605	59.1	6.10	3.619	.2913	.0806
K" 5	74	995	97.4	4.978	3.066	.2170	.0708
K' 3	74	597	58.6	6.07	3.654	.2955	.0810

flow loss are combined in Fig. 10, and we see that only Coils B and K are inferior to Coil A. The improvement over A is largest for Coil I, 15%. Another interesting feature is that only Coils I and J are superior to Coil A in both respects. They give more heat transfer and less flow loss per coil, i.e., per ft of tube length.

The effect of higher turbulence levels is also shown in Fig. 10. The 5.0% turbulence tests were made at approximately 600 fpm for coils B, G. H, I, J and K. Extrapolated results for each coil are shown as a point. Coil K, which is similar to a flat fin, in that it offers no boundary-layer relief, is little affected by higher turbulence. (This is an indirect corroboration of the contention that turbulence would not benefit the flat fin.) The group which are superior to Coil A (i.e., G, H, I and J) benefit by an average 4% at 600 fpm from the higher turbulence.

CONCLUSION

Surfaces having separated and unseparated flow have comparable heat transfer rates, but separation results in much higher flow losses. The effectiveness ratio, defined in this study, evaluates these effects and is a sensitive indicator of the performance of a surface. Visualization studies of models of cylinders and of finned tube exchangers indicated conditions which cause flow separation on various types of surfaces.

Fig. 8 Heat transfer effectiveness vs flow Reynolds number. *Undetermined low turbulence level for coil A

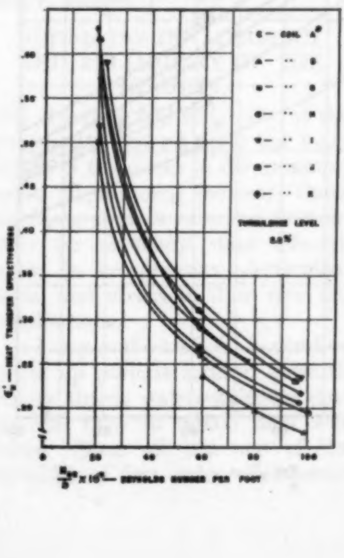
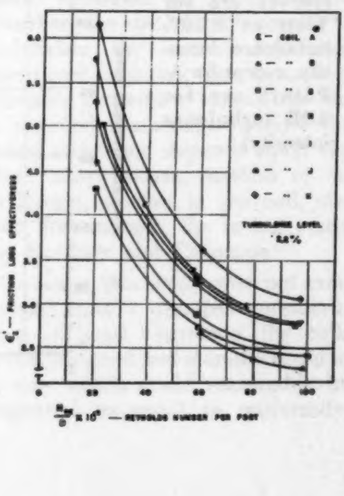


Fig. 9 Flow loss effectiveness vs flow Reynolds number *Undetermined low turbulence level for coil A



A relieved fin, which would provide boundary layer relief and avoid separation on the fin surface was designed. Coils, made up of such fins, were compared over the air velocity range 200 to 800 fpm with flat fin coils. The comparisons, made for single and two-row inline and staggered arrangements in a low turbulence stream, indicated that fin relief was excessive in this design.

Initial tests with the first relieved fin design indicated that the relieved fin benefitted from disturbances in the air stream. Since turbulence of a high intensity is often present in actual equipment, the relieved fin was tested at turbulence levels of approximately 2.2 and 5.0%. Turbulence substantially increased heat transfer and reduced the flow losses for this coil. The effectiveness ratio under these conditions somewhat exceeded that of the flat fin.

Five additional relieved fins were designed. They were tested as single coils in streams having turbulence levels of approximately 2.2% and 5.0% intensity. At 2.2% turbulence intensity, four of the five fins had a higher effectiveness ratio than the flat fin, and two of the fins had both a higher heat transfer rate and a lower flow loss. At 5.0% turbulence, the improvement was even greater.

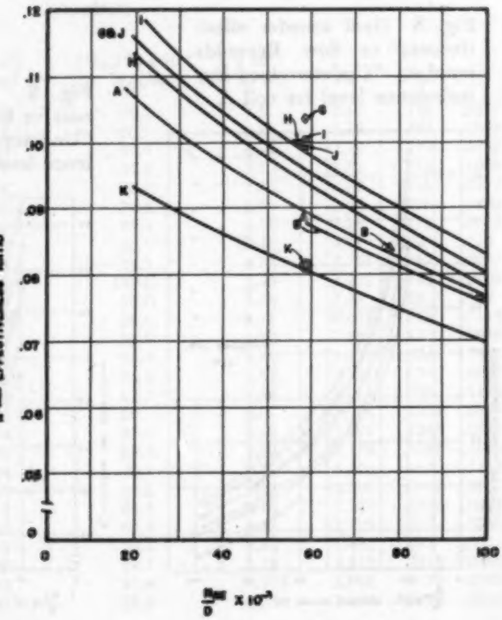


Fig. 10 Effectiveness ratio vs Reynolds number (curves are for Tests at 2.20% turbulence intensity, except for A. Points are for 5.0% turbulence intensity)

This series of observations and tests has shown that rational design, based upon flow considerations, can lead to much improved exchanger surface. The particular flat fin against which the comparisons were made is believed to have almost optimum thickness, width, and tube spacing for these flow conditions. Yet several of the second group of designs give more heat transfer with less flow loss.

Improvement was 15 to 20% for the best designs. However, it would be unrealistic to assume that this amount of improvement is a limit for optimum relieved fin designs in actual use in exchangers. The writer feels that considerably better designs are possible. Another important factor is that the fins tested in this study were designed as much for low flow loss as for high heat transfer. Greater improvement seems possible, if only one of these two characteristics is sought.

ESTIMATES OF ACCURACY AND RELIABILITY OF THE RESULTS

The accuracy and reliability of the test results depend upon the magnitude of the errors in the measurements taken during the tests. There are three sources of error: fluctuations (or unsteady state effects), errors in temperature determinations, and errors in flow rate determinations.

Long and carefully controlled warm up periods insured approximately steady state conditions during the time in which data was taken. Since all but one of the variables of first order effect were instantaneous readings, the period of a run was quite short. Observed drifts during the period of a run indicated an error of at most 1.0% in heat transfer rates.

Estimates of inaccuracies in the air side temperature difference and mass rate of flow determination indicate a maximum error in the computed heat flow rate of 2.0%. Similar estimates for the water side suggest a maximum error of 3.3%. Therefore, the max expected difference in the two heat flow rates for a given run is 5.3%. Max difference found was 4.0% and occurred in only one run. The average difference, for all runs, was 2.0%.

The principal question, concerning the reliability of the measurements as an indication of the relative performance of the various types of fins, arises in connection with the uniformity of the "bond resistance" between the tubes and fin collars for various coils tested.

Reliable measurement of this bond resistance is quite difficult. Further, the use of such results to subdivide the overall resistance measured in actual tests is a questionable procedure, unless one has a detailed knowledge of how the bond resistance depends upon the fluid temperatures, method of installation, stresses in the coil, etc. See, for example, the recent study by Gardner and Carnavos.[6]

Since this study was not concerned with a detailed consideration of bond resistance, the coils were designed and manufactured in a way which could reasonably be expected to result in uniformity.

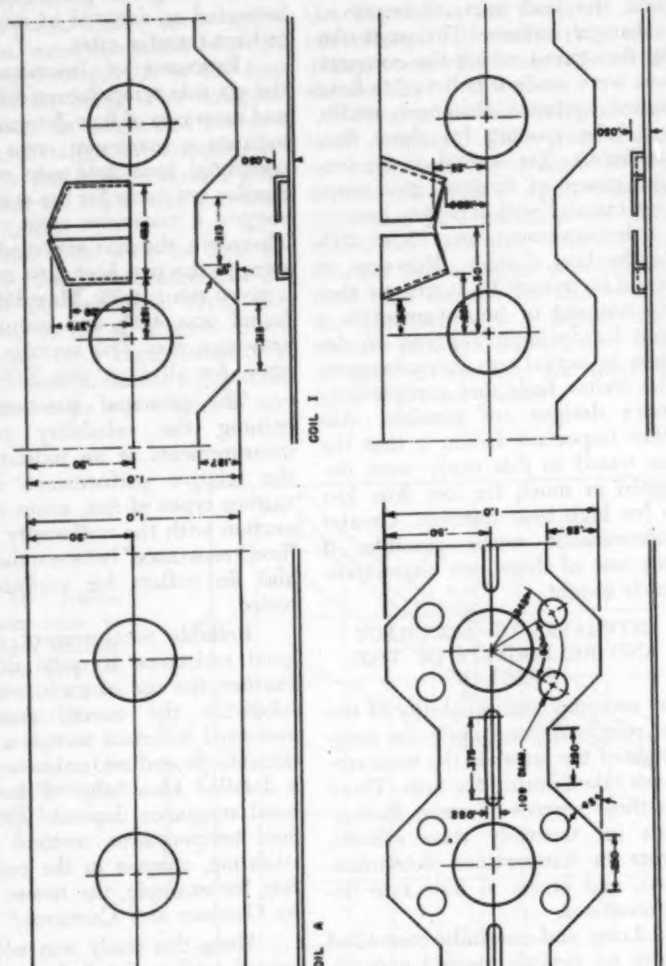
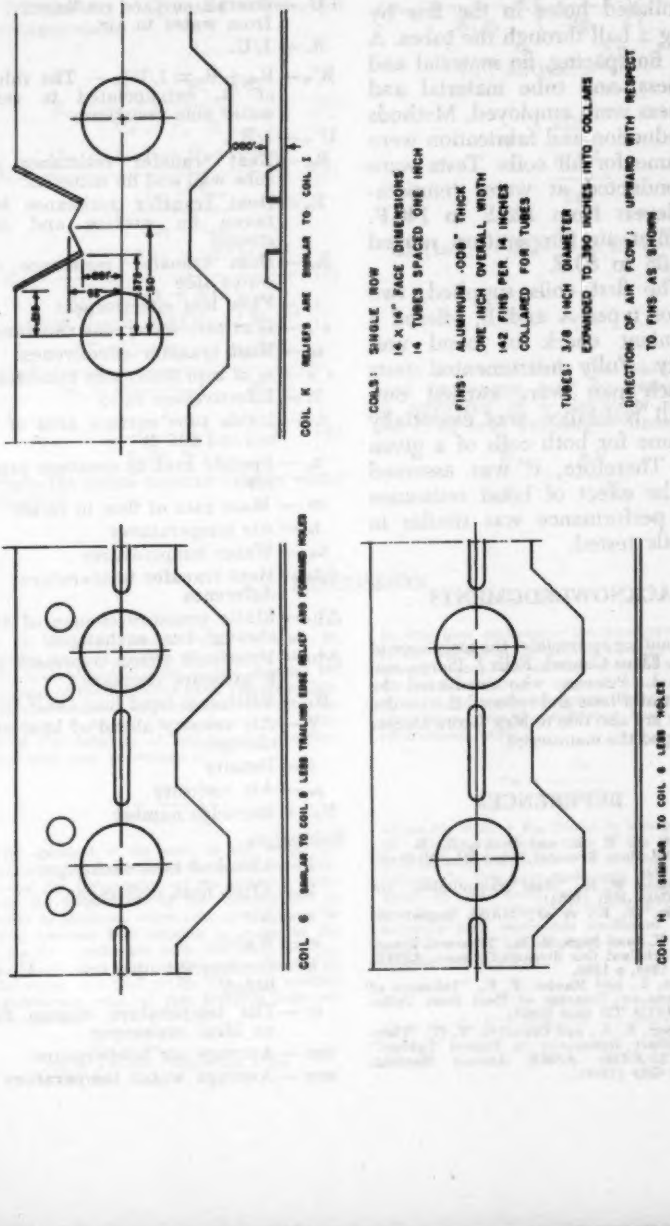


Fig. 11 Fin designs


COIL K RELIEFS ARE SIMILAR TO COIL I
.080"
COILS: SINGLE ROW
14 X 14" FACE DIMENSIONS
14 TUBES SPACED ONE INCH
FINS: ALUMINUM .008" THICK
ONE INCH OVERALL WIDTH
142 FINS PER 14 INCHES
COLLARED FOR TUBES
TUBES: 3/8 INCH DIAMETER
EXPANDED TO .390 INCHES INTO COLLARS
DIRECTION OF AIR FLOW: UPWARD WITH RESPECT
TO FINS AS SHOWN
TUBE INSIDE SURFACE AREA PER COIL — 1.496 FT²
COIL G SIMILAR TO COIL G LESS TRAILING EDGE RELIEF AND FORMING HOLES
COIL H SIMILAR TO COIL G LESS HOLES

One manufacturer supplied all coils, which were assembled by expanding the copper tubes into the collared holes in the fins by forcing a ball through the tubes. A single fin spacing, fin material and thickness, and tube material and thickness were employed. Methods of production and fabrication were the same for all coils. Tests were all conducted at water temperature levels from 120 F to 140 F. The inlet air temperature ranged from 65 to 80 F.

The first coils supplied, two each of types A and B, offered a convenient check of bond uniformity. Fully instrumented tests of each pair were carried out. Overall resistance was essentially the same for both coils of a given type. Therefore, it was assumed that the effect of bond resistance upon performance was similar in all coils tested.

ACKNOWLEDGMENTS

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REFERENCES

1. Eckert, E. R. G. and Drake, R. M. Jr., "Heat and Mass Transfer," 2nd Ed., McGraw-Hill (1959).
2. McAdams, W. H., "Heat Transmission," 3rd Ed., McGraw-Hill (1954).
3. Dryden, H. L., et al., NACA Report 581 (1937).
4. Sato, K., and Sage, B. H., "Thermal Transfer in Turbulent Gas Streams," Trans. ASME, Vol. 80, 1958, p 1380.
5. Kestin, J., and Maeder, P. F., "Influence of Turbulence on Transfer of Heat from Cylinders," NACA TN 4018 (1957).
6. Gardner, K. A., and Carnavos, T. C., "Thermal Contact Resistance in Finned Tubing," Paper 59-A-135, ASME Annual Meeting, Atlantic City (1959).

NOTATION

U_o — Overall surface coefficient from water to air.

R_o — $1/U_o$

R'_o — $R_m + R_s = 1/U'_o$ — The value of R_o extrapolated to zero water side resistance

U'_o — $1/R'_o$

R_m — Heat transfer resistance of tube wall and fin material

R_s — Heat transfer resistance between fin surface and air stream

R_w — Heat transfer resistance of water side

ϵ_F — Flow loss effectiveness

ϵ'_F — ϵ_F at zero water side resistance

ϵ_H — Heat transfer effectiveness

ϵ'_H — ϵ_H at zero water side resistance

N — Effectiveness ratio

A_i — Inside tube surface area of a coil $= 1.496\ ft^2$

c_p — Specific heat at constant pressure

m — Mass rate of flow in lb/sec

t_a — Air temperatures

t_w — Water temperatures

Δt_h — Heat transfer temperature difference

Δp_s — Static pressure change of air through heat exchanger

Δp_f — Frictional effects expressed as a pressure decrease

H_L — Frictional head loss (in.H_2O)

V — Air velocity ahead of heat exchanger

ρ — Density

μ_a — Air viscosity

N_{Re} — Reynolds number

Subscripts

1 — Ahead of heat exchanger

2 — After heat exchanger

a — Air

w — Water

h — Denotes Δt upon which U_o is based

m — The temperature change for an ideal exchanger

am — Average air temperature

wm — Average water temperature

APPENDIX

Calculation Procedures

Outline for single and double coil arrangements

$$R_o = \frac{1}{U_o} = \frac{A_t \; \Delta t_h}{q} \qquad (1)$$

$$\Delta t_h = \frac{(t_{w1} - t_{am}) - (t_{w2} - t_{am})}{\ln \dfrac{t_{w1} - t_{am}}{t_{w2} - t_{am}}} \qquad (2)$$

$$R_w = \frac{1}{h_w} = \frac{4.14 \times 10^{-4} \, m_w^{-0.8}}{(1 + .011\, t_{wm})} \qquad (3)$$

$$\text{from } N_{Nu} = .023 (N_{Re})^{.8} N_{Pr}^{1/3}$$

$$\epsilon'_H = \frac{U'_o \, A_t \, \Delta t'_h}{m_a \, c_p \, (t_{w1} - t_{a1})} \approx \frac{U'_o \, A_t}{m_a \, c_p} = \frac{A_t}{c_p \, m_a \, R'_o} \qquad (4)$$

where the prime denotes infinite water velocity

$$\Delta p_f = \Delta p_s - \Delta p_m \qquad (5)$$

$$\Delta p_s = \Delta H_m \, \rho_{01}$$

$$\Delta p_m = \frac{m \, \Delta V}{A_1 \, g_o} \approx \left(\frac{\rho_1 \, V_1^2}{2 g_o} \right) \left(\frac{2 \Delta t_a}{T_{a1}} \right)$$

$$H_1 \, ('H_2O) = \frac{\Delta p_f}{\rho_w}$$

$$\epsilon_H = \frac{\Delta p_f}{\dfrac{\rho_1 \, V_1^2}{2 g_o}}$$

$$H_L \, (\text{Iso.}) = H_L \frac{\rho_1}{\rho_m} = H_L \frac{T_{am}}{T_{a1}} \qquad (6)$$

$$N' = \frac{\epsilon'_H}{\epsilon'_B} \qquad (7)$$

Air flow conditions upstream of the heat exchanger may be generalized by referring to a Reynolds number per ft of heat exchanger dimension.

$$\frac{N_{Re}}{L} = \frac{\rho_a V}{\mu_a}$$

DISCUSSION

C. O. MACKEY, Ithaca, N. Y. (Written): In finding the heat transfer effectiveness ratio for the heat exchanger with a zero thermal resistance of the tube-side fluid, Professor Gebhart has made an approximation that may be replaced, if desired, by a more exact treatment. His definition of heat transfer effectiveness ratio may be written as

$$\epsilon_H = \frac{t_{a2} - t_{a1}}{t_{w1} - t_{a1}}$$

This quotient is the ratio of the actual increase in air temperature to the difference between the temperature of the entering water and the temperature of the entering air. In order to eliminate water-side performance, he then assumed heat transfer to occur in the same heat exchanger with the same t_{w1}, t_{a1}, m_a, A_t, C_p but with no thermal resistance on the water side ($R_w = 0$). The heat transfer effectiveness ratio of this fictitious heat exchanger is

$$\epsilon'_H = \frac{t'_{a2} - t_{a1}}{t_{w1} - t_{a1}} \quad \text{where } t'_{a2} > t_{a2}$$

In this heat exchanger, the temperature rise of the air would be greater than in the exchanger tested. The assumption made was that the m.t.d. in this new exchanger is $t_{w1} - t_{a1}$, which is not precisely correct. The use of this assumption gives the following heat transfer effectiveness ratio:

$$\epsilon'_H = \frac{A_t}{m_a \, C_p \, R'_o}$$

where $R'_o = R_a + R_m$ (found by extrapolation)

The rise in temperature of the air in the heat exchanger with $R_w = 0$ may be found as follows. In this heat exchanger, the temperature of the water may be assumed to remain at t_{w1}, because the flow rate for infinitely high water-side coefficient of heat transfer would be infinitely great. The rate of heat transfer is

$$q' = m_a \, C_p \, (t'_{a2} - t_{a1}) = \frac{A_t \, (t'_{a2} - t_{a1})}{R'_o \ln \dfrac{t_{w1} - t_{a1}}{t_{w1} - t'_{a2}}}$$

$$\text{Then } \frac{t_{w2} - t'_{a2}}{t_{w1} - t_{a1}} = e^{-\frac{A_t}{m_a C_p R'_o}}$$

$$\text{and } \frac{t'_{a2} - t_{a1}}{t_{w1} - t_{a1}} = \epsilon'_H = 1 - e^{-\frac{A_t}{m_a C_p R'_o}}$$

Only for very small values of $\frac{A_t}{m_a C_p R'_o}$ will ϵ'_H be equal to this ratio.

Air-conditioning engineers familiar with the equivalent by-pass concept will recognize in ϵ'_H an equivalent contact factor based on entering water temperature, for

$$t'_{a2} = \epsilon'_H t_{w1} + (1 - \epsilon'_H) t_{a1}$$

The effect of this correction on the reported results is to decrease ϵ'_H and N'. Some of the results given in Tables 11A and 11B are recalculated and included in this discussion.

Test	ϵ'_B	ϵ'_H	N'
A1	5.77	0.466	0.0808
A2	4.13	0.321	0.0777
A3	3.38	0.258	0.0762
A4	2.99	0.223	0.0745
B1	4.58	0.373	0.0814
B2	3.53	0.259	0.0734
B3	2.92	0.219	0.0750
B4	2.69	0.188	0.0698
B5	2.55	0.168	0.0658
B"3	2.69	0.210	0.0780
B"4	2.38	0.178	0.0748
B"5	2.28	0.158	0.0693
B'2	2.75	0.241	0.0877
B'3	2.69	0.210	0.0780
B'4	2.44	0.185	0.0758
B'5	2.28	0.160	0.0702
G"1	4.31	0.393	0.0911
G"3	2.747	0.230	0.0837
G"5	2.432	0.177	0.0728
G'3	2.70	0.245	0.0907
H"1	4.603	0.405	0.0879
H"3	2.881	0.237	0.0822
H"5	2.439	0.186	0.0762
H'3	2.721	0.237	0.0871
I'1	5.016	0.445	0.0887
I"3	3.301	0.278	0.0842
I"5	2.801	0.209	0.0745
I'3	3.175	0.273	0.0860
J"1	5.296	0.460	0.0869
J"3	3.251	0.267	0.0821
J"5	2.780	0.206	0.0740
J'3	3.239	0.269	0.0830
K"1	6.162	0.436	0.0708
K"3	3.619	0.252	0.0697
K"5	3.066	0.195	0.0636
K'3	3.654	0.256	0.0700

When the new values of effectiveness ratio, N', are plotted against N_{Re}/D, they are found to be lower than the values given in Fig. 10, and their slopes are smaller. Designs G, H, I, and J have the same effectiveness ratios, higher than A and B. Design K gives the poorest effectiveness ratio. Most of the values now fall between 0.064 and 0.090, instead of between 0.070 and 0.12 as shown in Fig. 10. The conclusions of the author about the relative merit of the designs is not modified appreciably.

Author Gebhart: This comment is a valuable addition to the paper in relating the equivalent by-pass concept and the equivalent contact factor to the quantities employed in this work. Looking at the heat transfer effectiveness from the point of view of a contact factor clarifies the concept. Some of the mystery associated with the flow loss effectiveness may be dissipated by recognizing it merely as a flow loss contact factor. The effectiveness ratio is merely a ratio of contact factors.

Professor Mackey has shown that the calculation procedure used in the paper amounted essentially to ignoring the higher terms in the expansion of an exponential. This approximation is unnecessary and should be avoided in future reductions of data. The differences between the reported results and the recalculated results, in the above table, are uniform. The difference increases with Reynolds number for a given fin design. This is because ϵ'_H, and, therefore, the error, both depend upon Reynolds number.

C. M. Ashley, Syracuse, N. Y. (Written): Unfortunately, the heat transfer effectiveness factor ϵ_H does not vary linearly with depth for the same type of surface since it approaches one for an infinite depth of surface. On the other hand, the flow loss effectiveness factor ϵ_B does vary substantially linearly with depth. Thus, the results of the comparison depend upon the depth of the coils used in the test as well as on the true heat transfer and flow resistance characteristics of the surfaces themselves.

It is recommended that the heat transfer effectiveness factor be changed by the omission of

$$\frac{\Delta t_h}{\Delta t_m}$$

This new factor expresses the heat transfer depth of the coil and is also equal to

$$\frac{t_{a1} - t_{a2}}{\Delta t_h}$$

The flow loss effectiveness factor ϵ_B is a useful criterion under the conditions where it is necessary to maintain the face area of the coil constant. However, in many situations, it is desired to determine the optimum type of heat transfer surface where the face area of the coil can be treated as a variable. Under these conditions, the use of Δ_{pf} taken alone as a flow loss factor is appropriate. The ratio of these two factors

$$\frac{\dfrac{\Delta p_f}{t_{a1} - t_{a2}}}{\Delta t_a}$$

then serves as a basis for coil optimization as shown in the paper, "A Proposed Method for Comparison of Effectiveness of Direct Heat Surfaces", ASHVE Transactions, 1925, A. E. Stacey, Jr. and C. M. Ashley. A further discussion and elaboration of this method is given in "Method of Comparing Heat Exchangers" by Donald G. Rich (ASHRAE JOURNAL, June 1960, pp. 50-52, 58).

In order to represent the same coil performance, a plot of heat transfer coefficient vs. the suggested effectiveness ratio is compared at the same values of effectiveness ratio. This basis of comparison emphasizes that a considerable increase in pressure drop can be tolerated for a reasonable increase of heat transfer. While I have not made a sufficient comparison to make certain that the rank order will change between other configurations, it is apparent that coil J is appreciably better than coil G on this basis of comparison and it is possible that other differences in rank order may show up on further analysis.

AUTHOR GEBHART: The magnitude of the differences between the performances of various coils may vary with heat exchanger depth but the direction of the differences will not vary. That the differences are not important in these results may be most simply seen as follows.

Professor Mackey's discussion shows that the approximation for ϵ'_H in the appendix of the paper may be replaced by an exponential. The first term in the expansion of this exponential represents the series for small values of ϵ'_H, i.e., for short exchangers, and this term is linear in A_1. The reported comparisons for the various fin designs were carried out in short exchangers, one row, and ϵ'_H was small. Therefore, since ϵ_H is more or less linear in A_1 the effectiveness ratio N′ is independent of depth (or of heat transfer rate) for these results.

Concerning the definition of ϵ'_H, I cannot agree that it should be altered as suggested. As defined in the paper, ϵ'_H has a fundamental physical significance. The fact that N′ is less for a deeper exchanger (having a uniform temperature fluid on the other side) merely shows that this is a poorer exchanger from the heat transfer–flow loss point of view. The downstream sections are equally effective for flow loss but less effective for heat transfer.

T. LEITERSDORF, Hamilton, Canada: Has viability of the different surfaces to fouling in service and the effect of this fouling on the effectiveness of the fins been considered?

AUTHOR GEBHART: The study outlined here did not attempt to consider this. The effectiveness ratio as defined in the paper is also an important function for surfaces in use. Fouling will decrease heat transfer. It will also increase friction loss, quite often by what amounts to essentially a separation mechanism. The effectiveness ratio would detect these effects.

W. A. SPOFFORD, Tyler, Tex.: It appears that in the mathematics the entering and leaving air temperatures were averaged and the arithmetic average was used. If the temperature differences were small, relative to the water temperature levels, this might be satisfactory; but if those temperature rises were large, this would lead into the refinements expressed by Mr. Mackey.

AUTHOR GEBHART: This is correct. The correlations are based upon the assumption that the air temperature difference is small compared to the difference in the air and water temperature levels. This is the case for the tests reported here. The water temperature was approximately 120 to 140 F, and the air temperature was room temperature, ranging from 65 to 80 F, on hot days.

1738

No. 1738

Field Laboratory for Heating Studies

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Current energy demands of the United States for power, processing, and heat average approximately 100,000 billion Btu per day, 30% of which is required for space heating. This proportion is likely to remain constant for the next century. But total energy requirements of the United States and the rest of the world are rapidly increasing both because of booming population growth and the impact of rapid technological changes on our standards of living. This expanded demand has resulted in marked increases in the costs of oil and gas during the past three decades; yet, an increase in the efficiency of conversion has resulted in reduction in the cost of electricity. All of these fuels are now economically feasible for space heating under some conditions. Nevertheless, the cost of all forms of space heating have risen, and the consuming public has become more conscious of the need for conserving space heating energy through adaptation of properly engineered heating systems coupled with thermally efficient structures.

The past three decades have seen calculation of heating loads progress from rule-of-thumb procedures used in the 1920's when fuels were cheap and space heating comfort demands not rigid, to the present time when the public demands close control of comfort conditions and engineering provides proper analytical procedures for load calculations and system design. Accurate laboratory tests

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for determining thermal and mass transfer characteristics of building materials now are used routinely.

But, despite these advances, many gaps remain in our knowledge of design and functioning of heating systems when applied to structures. Individual thermal conductivities of materials used in wall sections are known, but effects of framing members and other field installation variables are not duplicated in laboratory tests. Infiltration assumptions are approximate at best. For instance, tightly constructed and well insulated structures, in which infiltration is reduced to a minimum, may require special consideration as far as moisture and odor removal are concerned.

It appears probable that heating demands of a structure are, to an appreciable extent, dictated by ventilation needs and that in tight structures, either artificial ventilation or some internal means of moisture and odor removal may be required. The actual structure to be heated either rests upon or is sunk into a mass of earth of high heat capacity and relatively high thermal conductivity.

The structure to be heated is subjected periodically to a high intensity solar radiation field and a portion of this energy is absorbed on the outside surface of the structure and a portion is transmitted through fenestration to the interior. The outside of the structure is subjected to varying temperatures, varying wind conditions, rainfall and snowfall and solar radiation, and the inside to a variety of unscheduled heat and moisture gains. It is designed and heated that it may provide comfortable living conditions for people who live, work, and play inside; yet, we are not always certain of the effects of these living habits upon heating demands of the structure.

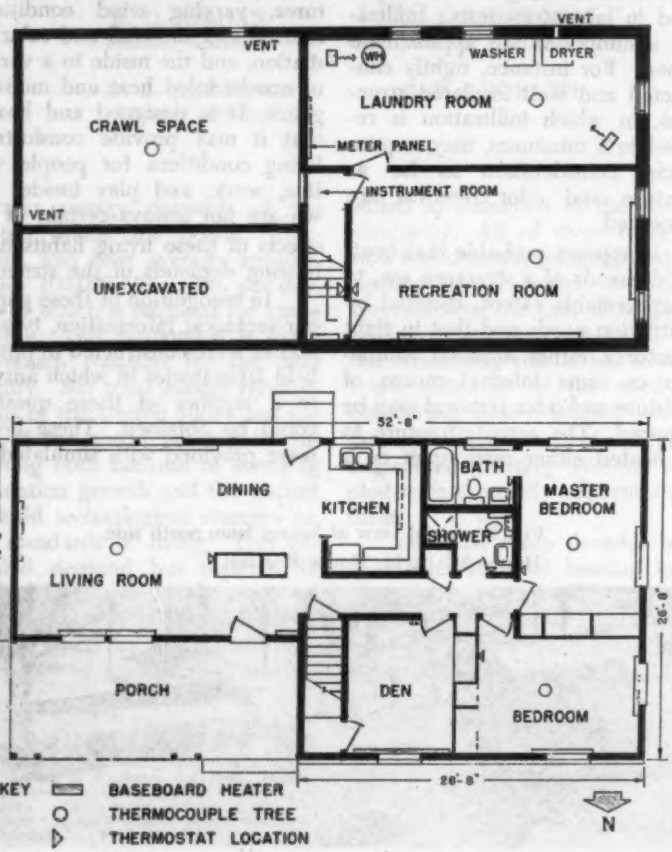
In recognition of these gaps in our technical information, two test houses were constructed to provide field laboratories in which answers to a number of these questions could be obtained. These houses were provided with simulated liv-

Fig. 1 Aerial view of houses from north side.
House A is right; House B is left

ing loads under accurate control and were fully instrumented to provide complete information on operating characteristics. A continuous record of the external environment was provided through a recording of weather conditions, including solar radiation. Soil analyses were made adjacent to the structures, and the earth's temperature variations recorded through a thermocouple well and adjacent to foundation wall.

Heating requirements of each room of the structures were monitored independently with separate

Fig. 2 Plan of first floor and basement showing location of baseboard heating units, thermostats, and thermocouple "trees"



electric resistance heaters zoned and metered to define the heating requirements for various areas. In short, an attempt was made to provide a complete picture of the interplay of various heating demands and heating losses to which a structure is subjected when placed in the transient environment in which it actually exists. This first article describes details of the structures, instrumentation, operating condition for the first winter's tests, and test objectives. Future articles will discuss the results obtained.

FIELD LABORATORIES AND THEIR ENVIRONMENT

Two test houses were constructed on adjacent 120 x 135 ft north-south oriented lots on the edge of Stillwater, Minn., 18 mi east of St. Paul. They are identical except for differences in insulation. Each was constructed with a system of panel wall components[1] with approximately 1100 sq ft of living area. Fig. 1 shows an aerial photograph of the houses and Fig. 2, a first floor and basement floor plan. The houses are north oriented. Both have approximately the same solar radiation exposure on the east, south, and west; shadow pat-

[1] Lu-Re-Co system of construction consists of preassembled modular 4 x 8 ft exterior wall, window and door panels (2 x 4 studs are 24 in. on center) and roof trusses paced 24 in. on center. Conventional exterior and interior finish materials are used to complete the house.

TABLE I

SUMMARY OF HEAT LOSS CALCULATIONS OF FIRST FLOOR AND BASEMENT ROOMS FOR ONE-HALF AND ONE AIR CHANGES PER HR AND THE INSTALLED HEATER CAPACITY FOR EACH ROOM

Room	Area Sq Ft	Calculated Btuh Heat Loss[a] House "A" Air Change 1	House "A" Air Change ½	House "B" Air Change 1	House "B" Air Change ½	Installed Heater Capacity Watts
Living Room[b]	448	16,498	13,552	17,071	14,126	4,400
Kitchen	106	2,992	2,293	3,112	2,413	800
Master Bedroom	160	5,294	4,249	5,529	4,485	1,600
Master Bath	42	1,386	1,113	1,451	1,178	500
NW Bedroom	150	5,045	4,059	5,321	4,335	1,600
Den	142	4,734	3,553	5,010	3,829	1,250
Shower	36	683	447	710	474	500
First Floor Sub Total[a]	1,084	36,633	29,266	38,204	30,840	10,850 W. 37,031 Btuh
Stair well[c]	40	695	695	695	695	1,000
Recreation Room[c]	329	546	546	570	570	3,750
Laundry[c]	288	2,072	2,072	2,140	2,140	4,000
Basement Sub Total[c]	657	3,313	3,313	3,405	3,405	8,750[d]
Total House	1,741	39,946	32,579	41,609	34,245	19,600 W. 66,895 Btuh

[a] Calculation based on double glazed windows and storm doors. For triple glazed windows, subtract 3419 Btu from total first floor heat loss. [b] Includes hall—68 sq ft.

[c] Basement heat loss calculated at ⅙ air change per hr and 50 F average inside air temperature.

[d] Basement heaters were oversized to provide rapid temperature recovery for intermittent heating.

tern photographs were taken at different times of the year to determine this point.

Table I presents a summary of sq footage of floor space, calculated design heat loss in Btu per hr, and wattage of electric heating elements installed in each room. All rooms on the first floor and the recreation rooms are heated by convection type baseboard electric units. Two 2000-watt blower type electric heating units were placed in the laundry areas of each house. During the early part of the first winter, line voltage thermostats were used in 7 different zones in each house. Later, two piece, low-voltage thermostats were installed in order to provide more precise temperature control needed in these comparisons. These units cycle the heating elements approximately 10 times per hr and provide temperature control within approximately ± 0.25 F.

House Construction and Heat Demand Calculations – Both test houses are identical with the exception of differences in type and thickness of wood fiber blanket insulation as shown in Table II. Over-all heat transmission coefficients as determined by calculation (corrected for framing heat loss) and guarded hot box test are included. Basement recreation room exterior walls in each house were insulated with ⅝-in. blanket between 2 x 2 in. furring and ½-in. insulating plank interior finish. The exterior walls in both laundry rooms were uninsulated.

Wall panels were constructed in the shop with 25/32-in. insulating board sheathing, 4 x 8 ft applied to 2 x 4 in. studs, 24 in. on center. Sheathing overlapped on panels ¾-in. at sides and bottom when panels were erected. A strip of sheathing about 12 in. in width was applied to the base of the wall to cover joists and sill plate. Exterior finish on east, west and south walls is insulating board shingle backer and prestained cedar shingles with 12 in. exposure.

Shingles on House A are buff color and on House B, light green,

TABLE II

SCHEDULE OF INSULATION USED IN EACH HOUSE WITH CALCULATED AND HOT BOX TEST "U" VALUES.

		Balsam-Wool Blanket Insulation		"U" Value	
House	Construction	Thickness	Liners	Calc.*	Hot Box Test
A	Ceiling	5 in.	Reflective	.049	.046
	Walls	3⅝ in.	Regular	.062	.057
	Floor	2 in.	Regular	.078	.082
B	Ceiling	3⅝ in.	Reflective	.060	.056
	Walls	2 in.	Reflective	.062	.066
	Floor	1 in.	Reflective	.063	.071

* Calculations corrected for heat loss through framing members (10% for ceiling and floor joists, 15% for wall studs, plates, and headers).

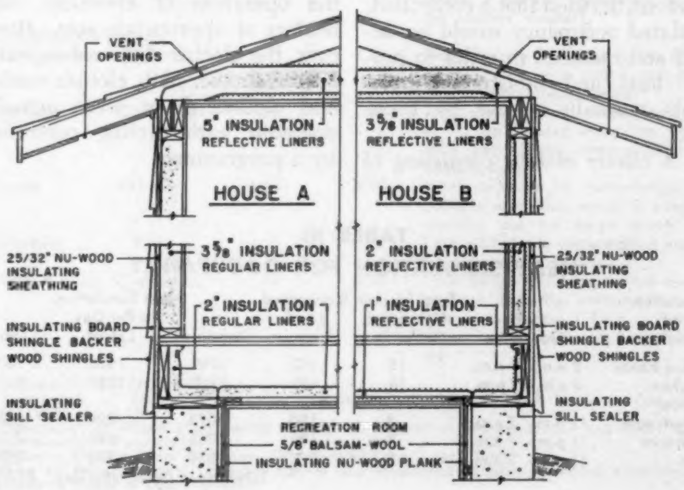
but laboratory tests, to determine equilibrium temperatures when exposed to sunlight, indicate practically no difference in solar absorption between the two colors. North wall of the two bedrooms is red face brick veneer and the two walls of the open porch are faced with ¼-in. hardboard panels, painted.

Between the masonry wall and sill plate, a wood fiber sill sealer blanket insulation was used. Joists are 2 x 10 in., spaced 16 in. between centers. The subfloor is ⅝-in. plywood with vinyl tile in the entrance hall, kitchen, bathroom, and shower room and ¾-in. oak in living room, dining room, and three bedrooms. After wall insulation was installed, ½-in. gypsum wallboard interior finish was applied.

Roof construction was ½-in. plywood on roof trusses placed 24 in. on center. There is a roof overhang of 24 in. from the wall and 7 screened openings in each of the north and south soffits which provide a total of 203 sq in. of unrestricted opening between the insulation and roof sheathing.

These are provided with covers in order that varying amounts of ventilation can be achieved. East and west gable walls have screened vents with a total free area of 318 sq in. House A is finished with dark brown asphalt roof shingles and House B, with dark green shingles. Again, solar absorption studies indicate very little difference between these colors. After the ceiling insulation was installed, 1 x 4 in. stripping was

Fig. 3 Cross section showing construction details and ceiling, wall and floor insulation in each house



nailed to the bottom of trusses, 16 in. on center, as a base for ⅜-in. gypsum wallboard interior ceiling finish.

All windows were weather-stripped wood sash with welded double glass. A removable third pane was installed from the exterior with clips. Window and door areas were approximately 15% of gross wall area. Fig. 3 presents a cross section drawing of the construction and insulation used in both houses. The living-dining room and three bedrooms were furnished with major pieces of furniture to occupy space and provide heat storage capacity.

Simulated Occupancy – Demands made upon a heating system are dependent to an appreciable extent upon living habits of the occupants of the space. For this reason, it is difficult to arrive at rational conclusions from field data. It was, therefore, decided that a controlled, simulated occupancy would be defined and installed in order to provide heat and moisture supplements normally added by occupants.

A family of four consisting of two males, both old enough to be considered adults, one adult female, and one child was postulated and heat and moisture gains to the space from these occupants based upon data available in the 1960 ASHRAE GUIDE. It was assumed that the family slept 8 hr in the house and averaged an additional 7 hr of more active life in the living space. Simulated loads, as shown in Table III, were actually distributed over a longer period of the day. Total heat supplied by the occupants was provided through electrical cone heating elements and a proportion of this heat translated to a moisture load by operation of a humidifier in which sensible to latent heat was translated by absorption of heat from the air.

Operation of refrigerator, range, freezer, television set, lights and miscellaneous electrical appliances were also simulated through the operation of electrical cone heaters of appropriate size. However, the electric dishwasher, water heater, shower bath, clothes washer and clothes dryer were actually operated with cycling controlled by a programmer.

TABLE III
HEAT SIMULATION FOR OCCUPANCY

Location of Simulator	Period of Operation	Total Heater Operation Hr Per Day	Connected Load kw	Heat Simulation Btu Per Day Sensible	Latent	Total
Living Room	8 a.m.-11 p.m.	15	.150	5759	1920	7679
Kitchen	8 a.m.-11 p.m.	15	.100	3840	1280	5120
Master Bedroom	11 p.m.- 7 a.m.	8	.150	3072	1024	4096
Bedroom	11 p.m.- 7 a.m.	8	.100	2048	682	2730
Den	11 p.m.- 7 a.m.	8	.075	1536	512	2048
				Total Heat Input, Btu/Day		21,673

Table IV presents a summary of simulated utility loads for this family of four persons as selected for these tests. The first column of this table shows the simulated or actual appliance load, and column 2 shows the period of use per day. Column 3 shows the actual connected load of the unit and column 4, the kilowatt hr per day for the actual or simulated appliance. Column 5 provides explanatory comments on the basis for simulation. References at the bottom of the table indicate the source material upon which these

TABLE IV
UTILITY LOAD SIMULATION FOR FAMILY OF FOUR

Simulation	Total Use Per Day	Connected Load kw	Total kw hr Consumption Per Day	Basis for Simulation
Actual Operations:				
Washer	50 min	.30	.25	7 complete wash cycles per week of 8# canvas cloths washed each cycle.[2, 5]
Dryer	55 min	4.24	3.90	7 complete drying cycles per week of 55 min per cycle. 8# wet canvas cloths taken from washer dried each cycle.[4, 5]
Dishwasher	50 min	1.02	.85	Average daily use estimated at one load per day.[2, 4, 5]
Water Heater	Automatic	2.60	14.16	Operates automatically to supply approximately 60 gal of 150 F water daily (37 gal simulating baths, 4 showers per day, lavatory use and losses; 7 gal for dishwasher; 16 gal for clothes washer).[1, 2, 3, 4, 5]
Simulated Operation:				
Refrigerator	12 hr	.10	1.20	Total daily kw hr consumption. Based on 50% operation and equals average monthly consumption.[3, 4]
Freezer	Continuous	.25	1.80	Total daily kw hr consumption equal to 1/4 hp freezer operating 1/3 of time and equals average monthly consumption.[2, 3, 4]
Range	100 min	2.00	3.40	Total daily kw hr consumption for 100-min operation equal to average monthly use for large cities.[2, 3, 4]
Television	6 hr	.15	.75	Total daily kw hr consumption equals 6 hrs average daily use.[4]
Lights & Misc	Continuous	.14	3.36	Total daily kw hr consumption equals average monthly consumption for misc. appliances, (e.g., toaster, mixer, radio, clock, etc.) and lighting.[4]

[1] ASHAE GUIDE 1959, Chapter 56
[2] Association of Edison Illuminating Companies, "Report of Load Research Committee 1957-1958", April 1959
[3] Association of Edison Illuminating Companies, "Report of the Residential and Rural Loads Subcommittee", June 1959
[4] Potomac Electric Power Company, "Elements of Load", April 1959
[5] Data on manufacturers' published figures and experience of sales personnel submitted by Minnesota Power & Light Company

assumptions were based.

Fig. 4 shows one of the simulated heat producing units located in the kitchen of the residence. This unit is composed of two cone heaters covered with protective hardware cloth and equipped with a relay controlled by the programmer, which also controls appliances. In this case, one of the heaters supplies the heat for lights and miscellaneous appliances and the other supplies the heat equivalent of the occupants while in the kitchen.

The heart of the load simulation equipment was the programmer shown in Fig. 5. This consists of 14 plastic cams with one cam representative of each activity. This was equipped with a 24-hr clock and the cams tripped microswitches which, in turn, operate the relays to power individual simulating heaters or actual appliances.

Fig. 4 Cone heaters and relay in kitchen simulating occupancy. Dishwasher operates 50 min daily at 7 p.m.

INSTRUMENTATION

Measurement of heating demands and inside and outside environment were provided by (a) an electrical power input measuring system, (b) a temperature measuring system, (c) a moisture measuring system, and (d) an external environment measuring system. Each will be discussed in turn.

Electrical Energy Input – Electrical energy consumption was recorded in each house by nine individual room meters and by two demand meters. The living room, kitchen, northwest bedroom, southwest bedroom, den, bath and shower on the first floor were metered individually and, in addition, meters were provided for basement laundry room and basement recreation room (including stair well). All individual room meters were calibrated to an accuracy of ±0.10%. Fig. 6 shows the location of individual room meters, demand meters, and electric clock. Not shown is camera equipment located in each house which photographed the meter panel board at 12 midnight each day. A timer was set to operate flood lights and through a 30-second delay bimetal

tube, the cameras recorded meter readings. During daily inspection of the houses, camera equipment was manually operated and reset for the following midnight photograph.

Temperature Measuring System — A total of 303 copper-constantan thermocouples were used to define temperatures inside and outside houses. Of these, 78 were recorded automatically every 24 min and 145 were read manually each week and were controlled by selector switches. An additional 80 thermocouples, installed to determine specific conditions, were read periodically when information was needed. All continuous recordings were made by means of two recording potentiometers and the manual readings, by means of a portable potentiometer connected to a manually actuated stepping switch.

Room temperature gradients were measured in the living room, southwest and northwest bedrooms, at the floor, three and 60 in. above the floor, three in. below the ceiling and at ceiling surface. In addition, recording resistance thermometers with a nickel bulb located near room thermostats provided continuous strip charts of room air temperatures in each house. All air thermocouples were shielded from radiation effects by means of concentric foil covered rings which permitted free air circulation but eliminated warm and cold radiation effects from window and wall surfaces.

Wall, ceiling and floor surface temperatures were measured at 22 locations. In addition, the temperature gradients through the north and south walls were measured at both the 60 in. level and across plate. Temperature gradients were also measured through the floor and ceiling constructions, both above and below the thermo-

Fig. 5 Programmer enclosed in plastics case located in laundry room which operated appliances and heaters simulating family of four in each house

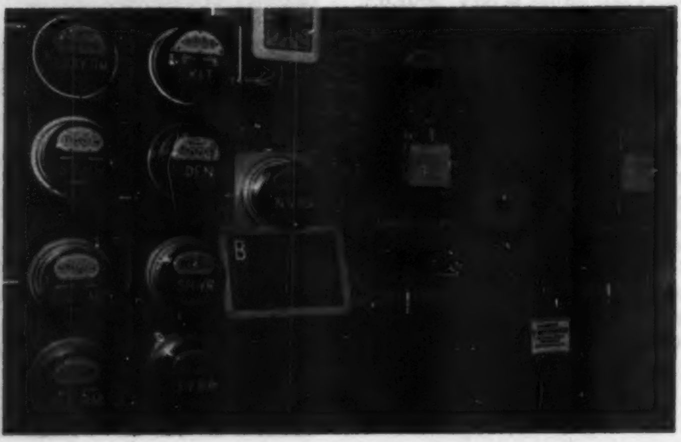
couple "trees" provided to measure room air temperatures. Both attic and crawl space air temperatures and surface temperatures were measured.

Ground temperatures are measured in the basement 2 ft, 1 ft and 4 in. below the surface as well as on the surface of the slab. Exterior block wall temperature was taken at 6 in., 2 ft, 4 ft and 6 ft depths outside the recreation room on the north and west walls and outside the laundry room on the west and south walls. In addition, soil temperature measurements were taken to a depth of 10 ft at a point 20 ft west of House B and 65 ft east of House A. These temperatures were recorded at the surface and at 1, 2, 3, 4, 5, 6, 8 and 10 ft depths. Complete soil analyses were made at the time thermocouples were installed.

Fig. 7 shows the instrument room located in the basement and provided as a central point for locating potentiometers and automatic selector switches for temperature programming equipment.

Moisture Measuring System—Continuous readings were made of relative humidity in the living spaces of both houses by two recording hygrometers. Relative humidities were checked weekly with a sling psychrometer in all rooms including attic and crawl spaces. In addition, periodic moisture measurements were made in the wall plates below the kitchen sink, shower room, northwest bedroom, and in the crawl space in the plate located above the concrete block. Moisture probes were permanently imbedded in plates at all measuring points and a Delmhorst mois-

Fig. 6 Central meter panel on east wall of laundry room has 9 individual meters and 2 demand meters



ture meter was used to record moisture contents.

External Weather Measuring System – A U.S. Weather Bureau type instrument shelter[1] was located midway between Houses A and B and contained a resistance type thermometer for air temperature measurements taken once every 2½ min. In addition, a weather instrumentation mast was located approximately 100 ft south of both houses to provide records taken at a 30 ft elevation of wind speed, wind direction and solar radiation. The solar radiation was measured by an Eppley 10 junction pyrheliometer connected to a millivolt recorder. Weather mast and air temperature instrument shelter are shown in Fig. 8.

[1] Model No. 176–Science Associates, Inc. Princeton, N. J.

OPERATION PROCEDURES

Although it was recognized that there was some merit in operating both houses under constant conditions throughout the winter, it was felt that it was more desirable to explore various operating combinations so that a wider variety of data might be obtained. For this reason, a varying schedule of operation was followed and for the period from October 31 to June 1, the houses were operated under conditions for full load simulation 37% of the time; under heat simulation, 15% of the time; and under conditions of no simulated load, 48% of the time. During 55% of the heating season, the houses

Fig. 7 Instrument room in basement contains 2 recording potentiometers and selector switches for reading temperatures with portable potentiometer

were operated with triple glazing on windows and for the remaining 45% of the season, with double glass.

For the initial two months of operation or approximately 25% of the heating season, houses were operated with the living room maintained at 72 F; baths, at 75 F; bedrooms, at 68 F; and stairway and basement, at 65 F. During this period, heat distribution characteristics were determined and following this, all rooms, with the exception of basement and stairway, were set at a constant room temperature of 70 F. The basement was heated to 65 F in two-hr cycles each day for a portion of the time, continuously on 24-hr cycles for another period, and no heat was supplied during a third period. A complete schedule of the different phases of operation together with an analysis of results will be presented in a subsequently to be prepared paper.

It was recognized that it would be impossible to artificially simulate infiltration through door openings and for this reason, a schedule of twelve door openings for entrance to or exit from the houses were provided manually. These openings were according to a fixed schedule and a logbook was provided to record these and all other manual activities to which the houses were exposed. During one period of operation, the door openings were increased to 24 per 24-hr period and during a few brief periods when inspection of the houses occurred, a much higher number of entrances and exits were recorded.

Here, particular care was taken to obtain an equal number of door openings and entrances to each house. It was recognized that the ventilation to which a house is exposed is, to a great extent, dictated by the living habits of the occupants, and that this varies markedly between families. In the present tests, door openings to which each house was exposed defined the simulated living habits for this particular simulated family.

Additional ventilation was provided by operation of the kitchen exhaust fan during the periods of

Fig. 8 View looking south showing U.S. Weather Bureau type instrument shelter and in background, 30-ft weather mast containing pyrheliometer, weather vane and anemometer

simulated cooking operations and the ventilation fans for the shower room during the periods when the showers were in actual operation. Further, it was recognized that under actual operating conditions when very high humidities are experienced, ventilation is usually provided by the occupants in order to alleviate these conditions. This ventilation is normally provided by additional door or window openings. In the simulated operation, a humidistat was installed to operate the central shower fan upon demand whenever predetermined excessive humidities were experienced. Doors extending from floor to ceiling permitted good circulation of air when the fan operated.

Each house was inspected daily and these inspections included checking operation of the programmer, circuit breakers, humidifiers, all appliances and all recording equipment. All electric meters were read and camera equipment used to record the midnight meter readings was checked and film advanced. All water consumption meters were read, and a load of clothes which had been washed in the electric washing machine shifted to the dryer. Drapes and shades in each room were inspected and any window condensation noted. Operation of dishwasher, range, fan, and the shower and shower fans were all checked.

The total amount of 1.2 gal of water was introduced daily into each house by the humidifier to simulate the calculated .96 and .24 gal water given off by occupants and cooking, respectively [1]. Daily consumption of all outside service meters were also read and snow depth, if any, adjacent to the house on all sides recorded. All vents were checked to make certain that they they were not obstructed. An exact schedule of operation was provided with each item to be accomplished in sequence in order that there would be no variations from day to day.

OBJECTIVES

The introduction presented the broad objectives which dictated the planning of this research program. More specifically, they may be subdivided as (a) heat loss studies, (b) temperature studies, and (c) moisture studies.

Heat Loss Studies – One of the principal objectives of the first winter's studies was to determine actual energy requirements and operating costs under a variety of defined and controlled inside conditions and a complete monitoring of the outside environment. These tests have permitted study of ceiling, wall and floor insulations and the effects of a variety of weather conditions including solar radiation and wind speed and direction upon the energy consumption of individual rooms as well as the complete houses. They have also provided a comparison of actual heat loss requirements with calculated heat losses.

Temperature Studies – Detailed temperature measurements have

[1] "Research in Humidity Control", Bulletin No. 106, Engineering Experiment Station, Purdue University, 1948.

permitted studies of room air temperature stratification and a comparison of wall temperature gradients with calculated gradients under different operating conditions. These studies include investigation of crawl space and attic air temperatures under different ventilation conditions and basement area and ground temperatures with seasonal variations.

Moisture Studies – Conducted to determine upper limits of indoor relative humidity as a function of the external enviornment, and field studies of effects of double versus triple glazing.

Future Studies–It is intended that these houses be operated during a second winter and that one house be occupied by a family of 4 people living under normal conditions. The second house will be operated under simulated conditions outlined previously for the entire heating season. This will permit a comparison of simulated conditions with actual living conditions and will permit a further evaluation of actual heat energy requirements of residential structures.

ACKNOWLEDGMENTS

This study was sponsored by Wood Conversion Company which wishes to acknowledge the valuable assistance and cooperation of Dr. R. C. Jordan, Professor and Head of the Mechanical Engineering Department, University of Minnesota, as technical consultant on the project. Also the Andersen Corp.; Minneapolis-Honeywell Regulator Company; Edwin L. Wiegand Co.; Northern States Power Co., all of whom have contributed liberally during the planning and executing of the work. The Minnesota Light and Power Co. also assisted in planning.

DISCUSSION

SIDNEY COATS, Niles, Mich.: What type of hot box and what size specimen was used? Was a typical sample of this section taken? Also, were there windows?

AUTHOR ERICKSON: The test equipment was a standard guarded hot box complying with ASTM requirements. The unit could be rotated to test ceiling, wall and floor panels constructed and insulated in the same manner as used in the test houses. Panels were 5½ ft wide and 5 ft high.

No window units were tested in the hot box. "U" value data used in the calculations were based on information received from the window glass manufacturer.

HERBERT ZIEL, Detroit, Mich.: Was the same frame with three glazings (single, double and triple) used or were there separate frames?

AUTHOR ERICKSON: All windows were weatherstripped wood sash with welded double glass. The third pane of glass was installed in the same sash from the exterior and held in place with clips.

ROBERT BOYD, Pittsburgh, Pa.: This study would appear to support strongly the thesis that insulation and double and triple glazing returned greater savings in capacity required and energy consumed for heating than conventional calculations indicate they will. It is evident that much valuable data have been recorded and a continuing flow of excellent information from correlation and analysis of this information is anticipated.

LESLIE BURY, Tulsa, Okla.: Why was the relative humidity maintained at 62% when the picture was taken? What was the relative humidity maintained throughout the study?

AUTHOR ERICKSON: The humidity was not controlled in the occupied House A during the second winter. The picture was taken in October and the humidity of 62% at that time was not harmful due to mild outside temperatures. During cold winter months, the humidity was in the range of 40-45%. When windows had double glass, condensation occurred in zero degree weather. There was no trouble with triple glazing.

T. WETHERINGTON, JR., St. Petersburg, Fla.: In the data collected on the demands, is there a record available on the diversity of the heating demands with the remainder of the house and, if so, does it indicate the heating requirements; or what percentage are the actual heating requirements as shown by these demands, a portion of the calculated demands for requirements for heating?

Also, is it planned to rotate the oc-

cupants between the houses in future tests to eliminate the differences in the way families operate and react to this heating system?

AUTHOR ERICKSON: It is difficult to answer the first question at this time. Detailed studies are being made of the heating demands under different external and internal conditions. This will be reported in the second technical paper.

As to rotating the families, this is not planned. The second heating season, House A is occupied by an actual family of four and House B is being operated with a simulated family. The project will be continued a third winter with two actual families occupying the houses.

F. E. WITTIG, Houghton, Mich.: Have any measurements been made of the moisture penetration in the walls?

AUTHOR ERICKSON: Moisture measurements are made periodically in the wall plates at four locations using a Delmhorst moisture meter. As I recall, the readings were comparable to average homes during winter months. These data will be included in a later report.

DONALD GOVAN, Cote St. Luc, Que.: Has any estimate been made of the heat loss in the water from the washing or dishwashing operation?

AUTHOR LEONARD: Yes, by metering the consumption of hot water and knowing its heat content, then measuring the amount of water used by the various appliances, we could calculate the amount of heat lost to drain water and also the heat which dissipated in the rooms. This information will be published in future articles.

LACEY BARNES, Owego, N. Y.: Has venting the dryer inside in the winter to gain that heat been considered?

AUTHOR ERICKSON: In well-insulated, tightly-constructed homes, clothes dryers should be ventilated to the outside, otherwise the humidity inside the house will be excessive and condensation problems may occur. It is true that most of the heat from the dryer is lost in the venting operation.

1739

No. 1739

Thermal Conductivity of Porous Materials

MARK E. STEPHENSON, JR. MELVIN MARK

The increasingly airborne aspect of our physical environment has emphasized the importance of reduced weight materials. These are often porous or cellular in construction. Greater use of such substances as plastic foams and honeycomb materials underscores this significantly. Fibrous insulations, textiles, granules and powders are all examples of materials which could be classified as cellular in the sense used here; that is, substances containing gaseous pores. For design purposes an important engineering property is the so-called thermal conductivity, especially since the application of these substances as insulation is very common. A knowledge of the functional relationships between thermal conductivity, cell size and shape, apparent density, and material-property values for cellular materials is useful, but the general problem is fairly complex, and a simple relationship applicable to all cases is not available; qualitatively a better understanding of the relationships can be obtained by examining simplified models representing the materials and correlating conclusions obtained from these models with the limited available experimental data.

MECHANISMS OF HEAT TRANSFER

Thermal conductivity of a cellular material is a complex property, since all three mechanisms of heat transmission, that is conduction, convection, and radiation, can play a role. Some heat is conducted through solid material and some through gas in pores. Convection also occurs in any gas pockets large enough to support circulation currents. Finally, radiation can travel from one surface to an-

Melvin Mark is Dean of Faculty, Lowell Technological Institute and a consulting engineer; Mark E. Stephenson was with the Raytheon Manufacturing Co. This paper was prepared for presentation at the ASHRAE Semiannual meeting in Chicago, Illinois, February 13-16, 1961.

other in each cell, and some radiation may even travel through the solid if it is partially transparent to infrared. It can be seen from this that "thermal conductivity" is not strictly a proper term. McAdams[1] recommends that results for cellular materials be expressed as conductances. However, since most of the available data is in terms of thermal conductivity, this will be used here, remembering that it means an overall apparent or effective thermal conductivity.

Since a cellular material possesses complex thermal patterns, it is natural that no simple single theory will suffice. The method of analysis used is generally adapted to the particular case being considered, certain less important factors being neglected compared to dominant ones, depending on permissible approximations for each case. However, since an approach suitable for one case is often unsuitable for another, assumptions involved in each analysis are of primary importance. Even for two substances composed of the same solid material in the same type of arrangement, a difference in cell size or density can change the relative importance of various mechanisms of heat transfer to the extent that the conductivity of one will fail to correlate with an analysis that fits the other.

METHODS OF ANALYSIS

In some applications radiation and convection can be neglected in comparison to conduction through solid material and gaseous pores. Upper and lower limits for effective conductivity of a two-component system can be calculated. The upper limit occurs with materials arranged in alternating plane layers parallel to the direction of heat flow. In this case

$$k_a = \phi k_s + (1 - \phi)\, k_g \qquad (1)$$

where k_a is the upper-limit effective conductivity of the cellular material, k_s and k_g are thermal conductivities of the solid and gas, and ϕ is the solid volume fraction. The lower limit, k_b, occurs with materials arranged in plane parallel layers perpendicular to the direction of heat flow. In this case

$$k_b = 1/[(\phi/k_s) + (1 - \phi)/k_g] \qquad (2)$$

Conductivity k of an actual cellular material might be expected to lie between k_a and k_b, the relative position between these limits depending on the physical arrangement of material. However, the ratio between k_a and k_b can be very great, of the order of ten to one or more, so that knowing only these limits is often of little practical help. An analysis of the problem for each particular type of geometrical arrangement of material considered is generally necessary.

Some relationships derived by Maxwell for electrical conductivity of mixed materials have been applied by Eucken[2] to thermal conductivity. For example, in the case of separated gaseous cells in a matrix of contingent solid material,

$$k = k_s \frac{1 - (1 - a k_g/k_s)\,(1 - \phi)}{1 + (a - 1)\,(1 - \phi)} \qquad (3)$$

where

$$a = 3/(2 + k_g/k_s) \quad (4)$$

The derivation requires that $(1-\phi)$ be small compared to unity, so this relation applies only to fairly dense cellular materials, being inapplicable to cases (for example, many foamed plastics) where the gas occupies most of the volume.

For fibrous materials in which the arrangement of fibers can be assumed to be approximately random, Schuhmeister[3] applied the following reasoning: Since the arrangement is random, the average fiber will have equal components projected on three perpendicular axes. Therefore, since two axes are perpendicular to, and one axis is parallel to, the direction of heat flow, a weighted average of the previously discussed upper and lower limits, k_a and k_b, might be expected in the form

$$k = \frac{1}{3} k_a + \frac{2}{3} k_b \quad (5)$$

Baxter[4] and Speakman and Chamberlain[5] managed to correlate some data on randomly arranged fibrous materials in this way, using numerical constants only slightly different from 1/3 and 2/3.

In the case of grains or powders, various physical models and the resulting analytical expressions have been proposed, for example, Deissler and Eian[6] and Schumann and Voss.[7] Deissler and Boegli[8] utilized a model consisting of spheres in packed cubical array and obtained a relationship for the effective conductivity k in terms of any given values of k_s and k_g. This gives a predicted position of k between k_a and k_b. Results check well with experimental data which Deissler and Boegli obtained on a number of various powders. This solution is for only one value of ϕ, $\phi = 0.52$. However, the contribution is valuable in that it takes into account, by a numerical relaxation technique, the fact that the heat flow lines are not unidirectional in a nonhomogenous substance.

The foregoing relationships express effective conductivity as a function of solid volume fraction. In fact, McAdams[1] states, in general, that "For nonhomogeneous solids the apparent thermal conductivity at a given temperature is a function of the apparent or bulk density ρ_a." Equations (1) and (2), as well as relationships such as Equation (3), predict an increase in k with increasing ϕ, as $dk/d\phi$ is positive for $k_s > k_g$. Since the apparent density ρ_a increases with solid volume fraction ϕ, these relationships predict an increase in k with increasing ρ_a. Therefore, for the same general type of physical arrangement of material, it may be expected that the k of an actual cellular substance might increase as ρ_a increases. This trend is reflected in many experimental results. For example, Harrington[9] presents data on the effective conductivity of a foamed plastic which shows $dk/d\rho_a$ positive everywhere in the range of ρ_a considered. In fact, Jacob[10] states, without qualification or statement of assumptions involved, that the effective conductivity of a loose or porous body "increases more than linearly

with the apparent density."

All the foregoing considerations would appear to lead to the conclusions that the effective conductivity of a cellular material (a) can be correlated as a function of apparent density, and (b) will increase with increasing apparent density. However, these two conclusions are not valid in general. The analyses discussed in the preceding are all based on the general approach of considering only pure conduction, neglecting any effects of radiation or convection. Presence of appreciable radiation or convection changes the situation considerably. Here importance of cell size should be emphasized. As used here, the term cell size refers to the average dimension of gas-filled pores. Apparent density is not a unique specification of the state of a cellular material, since it is possible to have more small cells, or fewer large cells, at the same overall apparent density. Gas cell size and apparent density must be considered as two independent variables, effective conductivity being a function of these two variables for any given type of substance.

The influence of cell size on radiation heat transfer is illustrated by the following simplified case. If two large black-body parallel flat plates, 1 and 2, perpendicular to the direction of heat flow, are at absolute temperatures T_1 and T_2 respectively, with $T_1 > T_2$, then heat will be transferred between them by radiation at the rate per unit area

$$q = \sigma (T_1^4 - T_2^4) \tag{6}$$

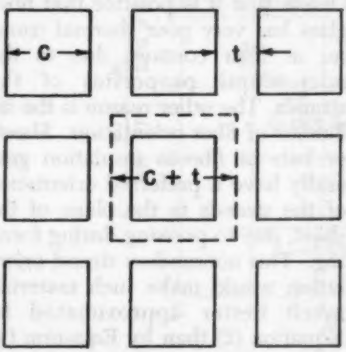
where σ is the Stefan-Boltzmann constant. Now assume a thin third black-body plate, at an intermediate temperature T_i, interposed between the other two. Then

$$q = \sigma (T_1^4 - T_i^4) = \sigma (T_i^4 - T_2^4) \tag{7}$$

and as $T_1 > T_i > T_2$, it is clear that the heat transfer has been reduced even though the interposed plate is of high emissivity and of negligible thickness or very high conductivity. Clearly, a larger number of cells per unit thickness through a material presents a greater impedance to radiation, as heat must be absorbed and reradiated more times.

Nusselt[11] made an analysis for cellular materials, assuming convection negligible and material arranged in alternate plane layers of solid and gas perpendicular to direction of heat flow. Although these simplifications limit the applicability of the results, the ap-

Fig. 1 Idealized cubical model of cellular material



proach is of interest since it includes the effects of radiation. When transformed into the same parameters and notation as used here, the result is

$$k = 1/\left\{ (\phi/k_s) + \frac{1}{[k_g/(1-\phi)] + (4\sigma T_{avg}^3/n)} \right\} \quad (8)$$

where T_{avg} is the average absolute temperature and n is the number of cells per unit length in the direction of heat flow. It will be noted that if radiation were neglected, Equation (8) reduces to Equation (2), as it should since the assumptions involved are then the same. Quantity n is significant in determining effects of radiation, and will be encountered again in the next section when certain general qualitative parameters are discussed.

Convection is influenced by pore size, since the effects of gas circulation due to natural convection will be greater in a larger cell. As used here, the phrase "heat transfer by convection" means any heat transmission by the gas over and above the lower limiting asymptotic value represented by pure conduction through stagnant gas. Allcut[12], who made conductivity measurements on many types of cellular materials, calculated percentage of heat transmitted by convection in his samples, but used a method which requires neglecting radiation. A significant parameter in convection is cell size c. This parameter will also be discussed in the next section.

An interesting study of fibrous materials, in particular, fiber glass, was made by Verschoor and Greebler.[13] They concluded that an almost negligibly small percentage of heat transfer takes place by solid conduction, the majority taking place by gas conduction, convection, and radiation. The relation assumed by Verschoor and Greebler can be obtained by taking Equation (2) and (a) neglecting ϕ/k_s, so that

$$k = k_g/(1-\phi) \quad (9)$$

(b) replacing this k_g with an equivalent sum of the effects of gas conduction, convection, and radiation, and (c) adding a correction term to account for what they found to be the "small amount of heat transfer due to the irregular contacts between the fibers." This contrasts with the work of Schuhmeister, whose Equation (5) takes into account the through-conduction described by Equation (1). However, there are two reasons why longitudinal through-conduction along strands of solid fiber, included in Equation (1), may be suppressed. One is that it is possible that fiber glass has very poor thermal transfer at fiber contacts due to the microscopic properties of the strands. The other reason is the influence of fiber orientation. Sheets or bats of fibrous insulation generally have a preferred orientation of the strands in the plane of the sheet, due to pressing during forming. This nonrandom strand orientation would make such materials much better approximated by Equation (2) than by Equation (1).

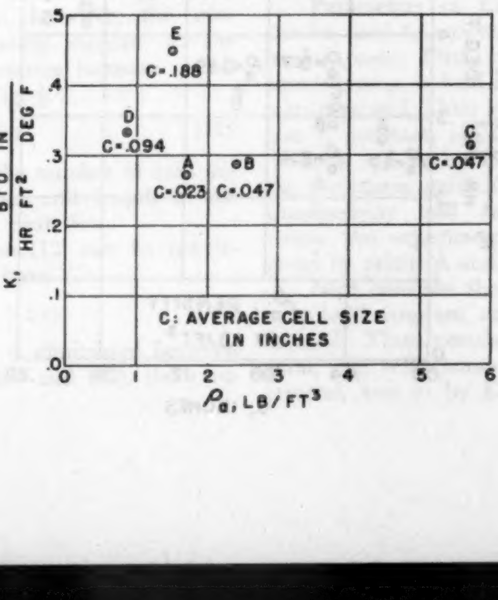
Fibrous insulating materials provide a good example of substances in which radiation and convection cannot be neglected. These materials often show effective conductivity decreasing with increasing apparent density, as was true over the entire range of apparent density considered by Verschoor and Greebler. However, other data shows effective conductivity of various fibrous materials, textiles, and so on, increasing with increasing apparent density. Foamed plastics show similar behavior; an increase in apparent density can produce either an increase or decrease in effective conductivity, depending on conditions. In fact, effective conductivity can go up or down with no change in apparent density. Additional independent variable cell size is here of paramount importance. Data is presented in the next section indicating that not only is apparent density alone an insufficient specification, but that it is often of less importance than the influence of cell size.

QUALITATIVE PARAMETERS

It can be seen from the foregoing discussion that large number of approaches to the general problem are possible. In order to guide the analysis of any particular case, certain qualitative parameters can be developed which aid in determining the importance of various modes of heat transmission, particularly how each mode will vary with changes in cell size and apparent density.

Consider a substance made up of a large number of randomly arranged cells (an example would be a foamed plastic). An idealized

Fig. 2 Thermal conductivity versus apparent density for polystyrene foam at 70 F



model of such a material can be taken to be as shown in Fig. 1. Cells are taken as cubes of side c, with wall thickness t between cells. By considering a typical block such as enclosed by the dashed lines, it follows that the density ratio will be

$$\rho_a/\rho_s = 1 - [1/(1 + t/c)^3] \qquad (10)$$

and the fraction of cross-sectional area for solid conduction will be

$$f = 1 - [1/(1 + t/c)^2] \qquad (11)$$

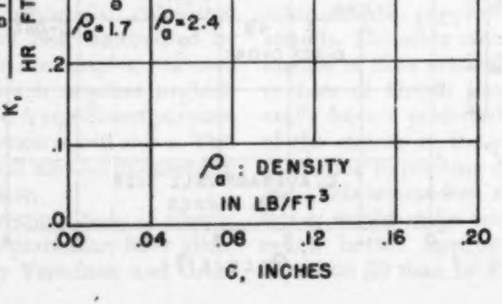
where ρ_a is the apparent overall density and ρ_s is the density of the solid material. In the following, ρ_s will be assumed constant. Clearly, ratio t/c, density ρ_a, and fraction representing solid conducting area f, all increase or decrease together.

Any excess of conductivity of material over the conductivity of gas in the pores must be due to additional contributions of solid conduction, radiation, and convection.

Heat transmitted by solid conduction is a function of f. As shown above, this increases with t/c. Instead of t/c, the ratio ρ_a/ρ_s can be considered, as both increase or decrease together.

Heat transmitted by radiation is a function of the reciprocal of the number of cells encountered per unit length, 1/n. This is because radiant heat flow is lowered when radiation cannot travel directly across a space, but instead must be absorbed and reradiated by a number of intervening walls as discussed earlier. The general

Fig. 3 Thermal conductivity versus average cell size for polystyrene foam at 70 F



phenomenon might be called a "cascade effect," and 1/n represents the freedom from cascade effect, or "directness" of heat flow.

Heat transmitted by convection is a function of two variables, cell size c, and 1/n. Larger cells can support more rapid circulating currents, while smaller cells restrict them and promote a stagnant condition. The cascade effect also exists for convection, since a stagnant film exists next to a solid boundary, so that each solid-gas interface creates a resistance to thermal transfer.

Effective conductivity can therefore be expected to increase with three parameters: ρ_a/ρ_s (representing solid conduction), 1/n (representing directness), and c (representing circulation). Of course, the conductivity is not linearly proportional to the above parameters, as these occur in the heat-transfer equations in complex way. For example, below a certain cell size (about 1/16 in.), circulation is almost nonexistent. At least qualitatively, however, the conductivity should increase as the above parameters increase.

From Fig. 1

$$n\,(c + t) = 1 \qquad (12)$$

where n is the number of cells encountered per unit length in the direction of heat flow.

Equation (12) can be rewritten in the form

$$1/n = c\,(1 + t/c) \qquad (13)$$

If $(1 + t/c)$ is eliminated between Equation (10) and (13), there results

$$\frac{1}{n} = c\,[1/(1 - \rho_a/\rho_s)^{1/n}] \qquad (14)$$

Equation (14) provides a simple relationship between the directness, circulation, and solid conduction parameters and emphasizes that conductivity is not a function of apparent density only, cell size being another important factor. This concept is supported by data obtained by McIntire and Kennedy [14]. They discussed improvement in insulating ability of polystyrene foam on going from a larger cell size, such as was made during World War II, to a smaller cell size similar to foams in use today. The data on conductivity given by McIntire and Kennedy are plotted in Fig. 2 as a function of density and Fig. 3 as a function of cell size. It can be seen that in the range of variables included in this data the conductivity is a stronger function of cell size than of density, since a single curve could be better fitted to the points of Fig. 3 than to those of Fig. 2.

Parameters of Equation (14) can be used to analyze some particular cases. First, consider the case where ρ_a is held constant and c is increased. Then solid conduction is constant, circulation is increased, and so by Equation (14) the directness must also increase. Conductivity will therefore increase. An experimental check is given by points A and E of Fig. 2.

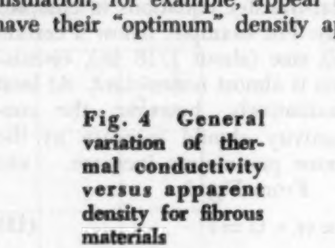
Next consider the case where c is held constant and ρ_a is increased. Then circulation is constant, the solid conduction is increased, and so by Equation (14)

directness must also increase. Conductivity will therefore increase. An experimental check is given by points B and C of Fig. 3.

Finally, consider the case where t is held constant and ρ_a is increased. If ρ_a is increased, the ratio t/c is increased, but t is constant, so c is decreased. Equation (14) or (12) then shows that 1/n is decreased. Therefore convection and radiation are decreased, but solid conduction is increased. From this it is not immediately clear whether over-all conductivity is increased or decreased. However, at very low values of ρ_a, where the cell size is large, convection and radiation are present to a large extent, while solid conduction is small. Hence an increase in ρ_a, which would decrease convection and radiation (although increasing solid conduction), would be decreasing the primary mechanism of heat flow. The conducductivity could, therefore, be expected to decrease. Conversely, at very high values of ρ_a, where cell size is small, the major contribution to heat transmission is by conduction, so that a reduction in this through a lowering of ρ_a could be expected to decrease the conductivity. Therefore, it might be presumed that a curve such as shown in Fig. 4 would result, with a minimum conductivity at some optimum value of density.

It would be difficult to obtain experimental data on foams corresponding exactly to constant t. However, consider the fibrous insulating materials such as mineral wool. Increasing the density is often achieved not by varying the diam of the strands, but simply by compressing the material. Such substances, of course, depart widely from a cubical model of cellular materials. However, some data, obtained by Finck [15] is shown in Fig. 5. The predicted behavior of thermal conductivity versus density checks with these results. This effect has also been noted by Wilkes[16] and more recently by Thigpen and Short.[20] As an application of this, if it is desired to obtain the max insulating effect from a fibrous material within a given available thickness of insulation, material should be used at its "optimum" density. This is often well above the density at which such fibrous insulating materials are commonly sold. In fact, some fibrous materials can be quite dense and still retain excellent insulating properties. Certain types of fiber glass insulation, for example, appear to have their "optimum" density ap-

Fig. 4 General variation of thermal conductivity versus apparent density for fibrous materials



proximately in the range of between ten and twenty lbs per cu ft.

A minimum conductivity at some "optimum" density may also be expected to exist in the case of some foamed plastics, since the above reasoning may well apply, at least qualitatively, even if the case is not one of exactly constant wall thickness. Some recently reported data support this conclusion; a curve of conductivity versus density given by Redman[17] is shown in Fig. 6.

INFLUENCE OF PROPERTY VALUES

Thermal conductivity of solid material will, of course, influence overall heat transmission. Even more important than thermal conductivity, however, may be the radiation characteristics of the solid material involved. Some heat is radiated directly across each cell from one side to the other. The emissivity of surfaces is therefore important. Also, some radiant heat may pass directly through thin cell walls. Radiant heat transfer will be reduced if both emissivity and transmissivity can be reduced, for example, by adding aluminum powder fill to a plastic foam. Such an addition would, of course, raise the thermal conductivity of solid material and so increase to some extent heat transmitted by solid conduction. The net effect may well be an improvement in insulating properties, however, if the fraction of cross-sectional area occupied by the solid material is fairly small, as is often the case.

The value of thermal conductivity of the solid material generally can be expected to be important when the solid conduction parameter is relatively large, while the emissivity of the solid material can be expected to be important when the directness parameter is relatively large.

Properties of gas filling the cell pores also influence over-all heat transmission. This gas need not necessarily be air. If it is of high molecular weight, its conductivity will generally be low. Its free-convection heat transfer would

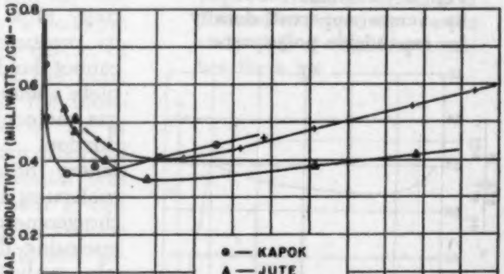
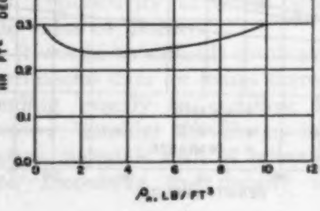


Fig. 5 Thermal conductivity (milliwatts/cm deg C) versus apparent density (gm/cm³) for various fibrous materials

be somewhat greater than air, largely because its density is greater, if space were sufficient to permit circulation. However, since convection is negligible in small cell spaces, the net effect is generally a lower heat transmission than with air. Considerable interest in insulation of this type has arisen in connection with construction of commercial refrigerators. With insulation of lower thermal conductivity, thickness can be reduced and food storage area made larger for the same external refrigerator dimensions. One problem, however, is diffusion of heavy gas out of, and air into, the insulation. A low-permeability "skin" must surround and seal insulation of this type.

It might be expected that percentage improvement in insulating effect through use of a heavy gas would be greatest when the circulation, directness, and solid conduction parameters are all small; that is, when heat transfer is primarily by gas conduction. In the case of a fibrous material of constant strand diam the following reasoning applies. At high density, heat transfer, largely through stagnant gas conduction, will be reduced, but at low density where large pores can support some circulation, heavy gas will provide greater heat transfer than air. Therefore, it seems reasonable to predict that with a heavy gas "optimum" density for a fibrous material will be shifted toward somewhat higher values of apparent density.

Fig. 6 Thermal conductivity versus apparent density for expandable polystyrene



MEAN-FREE-PATH EFFECT

Another factor connected with cell size is utilization of molecular "mean-free-path effect," making it possible to lower thermal conductivity of material below the thermal conductivity of the gas in the pores. Since the early work of Smoluchowski[18] and others, this has been a subject of interest. Its application to the effect of cell size on effective conductivity can best be introduced as follows:

First-approximation analysis of gaseous heat conduction by kinetic theory shows that thermal conductivity of a gas is independent of its pressure. But this conclusion cannot hold true down to a complete vacuum, since absence of all gas molecules precludes any conduction. At very low pressures, where the mean free path of gas molecules becomes comparable to the geometrical dimension of space enclosing the gas, the above simple analysis (involving statistical assumptions requiring large numbers of molecules, colliding much

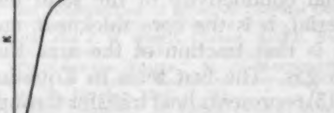
more often with each other than with the walls) breaks down. According to Jakob,[10] such pressures are of the order of about 0.05 mm Hg for enclosing-space dimensions commonly encountered. Further analysis shows that at extremely low pressures, approaching a pure vacuum, thermal conductivity is linearly proportional to pressure, approaching zero conductivity at zero density. Glass vacuum bottles, for example, are in this range.

A plot of thermal conductivity k, versus pressure p, for a gas would appear as shown schematically in Fig. 7. However, as mentioned before, thermal conductivity actually begins to drop off at pressures where mean free path is comparable to enclosing-space dimensions. The abscissa p in Fig. 7 is therefore not the proper independent variable; thermal conductivity should be plotted as a function of the ratio of mean free path, λ, to enclosing-space dimension, c, that is, λ/c. This has the form of a Knudsen number, Kn. Plotting the thermal conductivity against 1/Kn, that is c/λ, gives a curve of the same shape but emphasizes the fact that k is actually a function of the ratio c/λ. If cell size, c, of a cellular material is of the same order of magnitude as the mean free path, λ, possessed by air at sea-level pressure, then thermal conductivity of the material will enter the region in which it can be decreased below that of still air even though pressure is not reduced. But this requires extremely small pores, of the order of millionths of an inch. However, a few materials have been produced which do enter the range of this mean free path effect, and possess thermal conductivities less than that of still air. Examples are "Santocel" brand silica aerogel[19,20] powdered insulation, and "Min-K" brand molded insulation. Recent contributions to the study of this effect include Deissler and Boegli[8] for powders, and Verschoor and Greebler[13] for fibrous aggregates.

HONEYCOMB MATERIALS

A honeycomb material consists of a hexagonal cell array of core substance separating two facing sheets or skins. An analysis of the apparent conductivity of honeycomb materials has been made by Dunkle, Gier and Bevans.[21] They conclude that thermal resistance of the air cell is so large compared to that of solid material that it may be neglected, and the situation treated as two purely conductive resistances in series, that of the solid material in the core and that

Fig. 7 General variation of thermal conductivity versus reciprocal of Knudsen number for a gas



of the bond (which cements core to skin). Dunkle, Gier and Bevans succeeded in correlating experimental results for a large number of honeycomb specimens based on this approach. However, material property values, as well as the solid volume fraction, are here of considerable importance. Their samples were all of aluminum, which has a high conductivity. Neglecting heat transfer through the air cell is therefore a good approximation, as solid conduction naturally predominates. However, many honeycombs are made of laminated, resin-impregnated paper or glass materials, with much lower conductivities. In addition, these substances often have surfaces of high emissivity, quite contrary to the shiny surfaces so often associated with aluminum, so that radiation across the cell cannot be readily neglected. The correlation of Dunkle, Gier and Bevans cannot, therefore, be accurately applied to honeycombs of this type.

Mark[22] suggests an approximate approach for honeycomb materials in general. He writes for the effective conductivity of the core,

$$k = xbh/2 + (1 - x)\, k_s \qquad (15)$$

where h is the film coefficient for the honeycomb cell, k_s is the thermal conductivity of the solid material, b is the core thickness, and x is that fraction of the area that is gas. The first term in Equation (15) represents heat transfer through the gas cell part of the honeycomb, which is accomplished by convection currents, gas conduction, and radiation. Since there is a resistance to heat flow at each solid-gas interface bounding the cell, the term is composed of the sum of two individual surface resistances 1/h, or 2/h. It is multiplied by x to account for the proportional part of the area affected.

The value of h in Equation (15) is a function of core thickness b, cell size, temperature pattern, and cell wall properties. The value of h could be expected to lie between two limiting values. The lower limit would be the h-value equivalent to conductivity of a gas space equal to core thickness (with no radiation or convection present). The upper limit would be the equivalent value for a gas space infinite in two dimensions[25] but of thickness equal to the core thickness, convection and radiation being present. Mark suggests that, as a first approximation, a simple means of these two limits be used, and presents experimental data on plastic honeycombs showing good correlation with this approach.

Considering independent variable cell size, heat transfer by both convection and radiation is reduced when cell size is reduced. The reduction in convection is due to the small space which limits gas circulation. The reduction in radiation can be seen by examining the radiation equation.

$$q = \sigma F\, (T_1^4 - T_2^4) \qquad (16)$$

where F is a factor depending on the emissivities of surfaces involved and the geometrical configuration or shape. A honeycomb is basically different from foamed, fibrous, or

granulated materials in that only one cell exists from one side to the other. The "directness" for radiation as previously defined; i.e., 1/n, is therefore the same for all honeycombs. However, the radiation is influenced by the geometrical aspect ratio of cells (ratio of cell thickness to equivalent cell diam). For the gas cell of a honeycomb core, F depends upon some equivalent cell diam, cell length, and emissivity of cell walls. Specifically, F is proportional to the equivalent cell diam to cell length ratio,[1] cell length being the core thickness. Consequently, reducing the cell size reduces F, indicating a reduction in heat transferred by radiation across the core.

If the cell size is reduced by increasing the number of cells rather than by increasing thickness of the cell walls, and if x in Equation (15) is not decreased appreciably, then as the cell size decreases the conductivity of a honeycomb core could be expected to decrease. This has also been predicted elsewhere.[24] However, as the cell size is decreased, x in Equation (15) may decrease appreciably. Then reducing the cell size would be equivalent to replacing conduction through gas with conduction through the solid core material. Since the conductivity of solid-core material is normally greater than that of gas, conductivity of the core might increase; in other words, the increase in heat transferred by conduction might more than balance the reduction in radiation and convection. Some data exist which substantiates this.[21,25] As neither extensive test data nor exactly appropriate theoretical equations are available at the present time, it is not possible to draw a general conclusion on the effect of decreasing honeycomb cell size on effective conductivity.

Comparing the honeycomb core with a material such as a foamed plastic, it could be expected that the honeycomb would not be as good an insulator, since the plastic foam is made up of a large number of very small air cells (i.e., less directness and practically no circulation). Consequently, heat is transferred largely by conduction and radiation through the still gas and the plastic walls in the case of the foamed plastic, whereas convection, as well as direct radiation and often greater solid conduction, may be present to a larger extent in the honeycomb core.

REFERENCES

1. "Heat Transmission," W. H. McAdams, (McGraw-Hill, 3rd edition, 1954).
2. A. Eucken, (Forschung, a.d. Geb. d. Ingenieurives, Vol. 11, No. 6, 1940).
3. Schuhmeister, Ber. K., (Akad. Wien (Math-Naturrur Klasse), Vol. 76, 1877, p. 283).
4. "The Thermal Conductivity of Textiles," S. Baxter, (Proc. Phys. Soc., Vol. 58, 1946, p. 105).
5. "The Thermal Conductivity of Textile Materials and Fabrics," J. B. Speakman and N. H. Chamberlain, (J. Text, Inst., Vol. 21, 1930, p. T29).
6. "Investigation of Effective Thermal Conductivities of Powders," R. G. Deissler and C. S. Eian, (NACA RM E52C05, 1952).
7. "Heat Flow through Granulated Material," T. E. W. Schumann and V. Voss, (Fuel in Science and Practice, Vol. 13, No. 8, 1934, pp. 249-256).
8. "An Investigation of Effective Thermal Conductivity of Powders in Various Gases," R. G. Deissler and J. S. Boegli, (Trans. ASME, Vol. 80, No. 7 October, 1958, pp. 1417-1425).
9. "Polyurethane Foams," C. J. Harrington, (Society of Plastics Engineers Journal, Vol. 12, No. 10, October, 1956, p. 19).
10. "Heat Transfer," M. Jakob, (Vol. 1, Wiley, 1949).
11. W. Nusselt, (Zeitschr. d. bayer. Revisionsver, Nos. 13 and 14, 1913).
12. "An Analysis of Heat Transfer Through Thermal Insulating Materials," E. A. Allcut,

(Proceedings of the General Discussion on Heat Transfer, Institution of Mechanical Engineers, London, England, 1951, pp. 232-235).
13. "Heat Transfer by Gas Conduction and Radiation in Fibrous Insulations," J. D. Verschoor and P. Greebler, (Trans. ASME, August, 1952, pp. 961-968).
14. "Styrofoam for Low-Temperature Insulation," O. R. McIntire and R. N. Kennedy, (Chemical Engineering Progress, Vol. 44, No. 9, September, 1948, p. 727).
15. "Mechanism of Heat Flow in Fibrous Materials," J. L. Finck, (Bureau of Standards Journal of Research, Scientific Paper 243, Vol. 5, 1940, p. 973).
16. "Heat Insulation," G. B. Wilkes, (Wiley, 1950).
17. "Expandable Polystyrene. . . . Applications and Fabrications," D. F. Redman, (ASME paper 58-SA-65).
18. Report about the Second International Congress on Heat, (Vol. 2, Wien, 1910, p. 166).
19. "Thermal Conductivity of Silica Aerogel," Kestler and Caldwell, (Industrial and Engineering Chemistry, Vol. 26, No. 658, 1934).
20. "The Relation Between Heat Conductivity and Structure in Silica Aerogel," S. S. Kistler, (Journal of Physical Chemistry, Vol. 39, 1935, p. 79).
21. "The Thermal Conductivity of Honeycomb Material Near Room Temperature," R. V. Dunkle, J. T. Gier, and G. T. Bevans, (ASME paper 58-SA-1).
22. "A Design Note on Honeycomb Sandwich Construction," M. Mark, (Proceedings of the Ohio State University—WADC Radome Symposium, Vol. 1, August, 1956, pp. 51-56).
23. Thermal Resistance of Air Spaces," F. B. Rowley and A. B. Algren, (A. S. H. V. E. Transactions, Vol. 35, 1929, p. 165).
24. "Thermal Insulating Properties and Characteristics of Low Density Plastic Core Materials," A. C. Rawuka, presented at the Society of Aircraft Materials and Process Engineers Conference on Adhesive Bonded Structures for Aircraft, January 31, 1957, Los Angeles, California.
25. Research and Development Report No. 133, Hexcel Products Inc.
26. "The Apparent Thermal Conductivity of Fibrous Materials," J. J. Thigpen and B. E. Short, (ASME paper 59-A-293).

NOMENCLATURE

a = factor defined by Equation (4), dimensionless
b = core thickness of honeycomb material, in.
c = cell size, in.
F = factor in radiation equation, dimensionless
h = film coefficient of skin in honeycomb cell, Btu/hr ft^2 deg F
k = thermal conductivity of cellular material, Btu in/hr ft^2 deg F
k_a = upper-limit thermal conductivity of two-component mixture, Btu in/hr ft^2 deg F
k_b = lower-limit thermal conductivity of two-component mixture, Btu in/hr ft^2 deg F
k_g = thermal conductivity of gas, Btu in/hr ft^2 deg F
k_s = thermal conductivity of solid material, Btu in/hr ft^2 deg F
n = number of cells per unit length parallel to heat flow, in.$^{-1}$
p = pressure, mm Hg abs
q = heat flow rate per unit area, Btu/hr ft^2
t = wall thickness, in.
T_{av} = average temperature in cellular material, deg R
T_1 = temperature of intermediate flat plate, deg R
T_2 = temperature of warmer flat plate, deg R
T_3 = temperature of cooler flat plate, deg R
x = fraction of honeycomb area occupied by gas, dimensionless
λ = mean free path of gas molecules, in.
ρ_a = apparent over-all density, pcf
ρ_s = density of solid material, pcf
σ = Stefan-Boltzmann constant, Btu/hr ft^2 (deg R)^4
ϕ = solid volume fraction, dimensionless

1740

No. 1740

Heat Transfer through Mineral Wool Insulation in Combination with Reflective Surfaces

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Transfer of heat through air spaces used in building construction takes place by radiation, conduction and convection. Under certain conditions of temperature and direction of heat flow, resulting from the reduction in convective heat, the transfer of heat by radiation may be a large part of the total. To increase the thermal resistance of insulated structures, there has been a trend in recent years to combine the reflective surfaces with mineral wool insulation. This has introduced certain questions:

1. How accurate are conventional calculations of heat gains and losses through insulated frame structures?

2. How effective are reflective materials when used with mineral wool insulation under different conditions of heat transfer?

Experimental studies conducted in the past have provided some information. Present methods of calculation assume that the temperature gradient through framing members is linear and independent of adjacent air spaces and insulating materials. These methods, disregarding heat transfer to joist interfaces, also assume that heat transferred within stud or joist spaces is between two infinite parallel surfaces. Previously it has been assumed that temperature varied linearly along enclosed joist surfaces when the direction of heat transfer is downward through a joist space, and that there is negligible natural convection of air. Heat transfer is then primarily by radiation and conduction.

C. E. Lund is Professor and R. M. Lander is Research Associate, Department of Mechanical Engineering, University of Minnesota. **This paper has been prepared for presentation at the ASHRAE Semiannual Meeting in Chicago, Ill., February 13-16, 1961.**

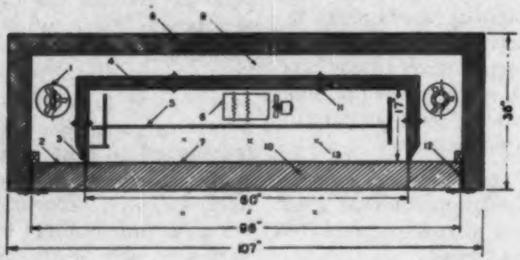
A theoretical study[1] has been made of heat transfer downward through a typical joist space and compared with some experimental data. It was found that radiation between the surfaces surrounding air space creates convection currents because of temperature variation along the joist interface, which deviates from the linear variation. The modified calculated values agree quite closely with experimental values as compared with present methods of calculation. However, to obtain the heat transfer by this new method involves a series of complicated and laborious calculations involving some intricate equations.

Guarded Hot Box – Thermal conductances of built-up panels representing walls, ceilings, and floors of residential construction for both summer and winter conditions were measured by a guarded hot box apparatus. It was also used on a pitched-roof-ceiling combination for winter test condition. Its design complies in all essential details with ASTM Designation C236-54T: "Tentative Method of Test for Thermal Conductance and Transmittance of Built-Up Sections by Means of the Guarded Hot Box." In Fig. 1, the guarded hot box is mounted on horizontal trunnions so the test panel can be rotated in any desired plane. The panel is placed within the outer or guard box and held rigidly against the heat metering box, using a rubber gasket as a periphery seal. The apparatus is located in a room which can be maintained at temperatures from 10 to 100 F and controlled to within 0.1 F.

Thermal conductances of test

[1] Downward Heat Transfer Through a Joist Space. Chung and Lund. (Heating, Piping and Air Conditioning, December 1958.)

Fig. 1 Tilting guarded hot box apparatus



1 FAN AND HEATERS
2 GUARD AREA
3 RUBBER GASKET
4 HEAT METERING BOX
5 BAFFLE
6 FAN AND HEATERS
7 TEST AREA 32" X 60"
8 GUARD BOX
9 GUARD SPACE
10 TEST PANEL 5'-6" X 8'-0"
11 DIFFERENTIAL THERMOCOUPLES
12 HAIR-FELT SEAL
13 24 GUAGE THERMOCOUPLES

panels are determined for a particular set of warm and cold side air temperatures by measuring the steady state heat flow rate through a panel. Twenty differential thermocouples, located in the center of equal areas, are connected in series to form a thermopile which indicates the average temperature difference, and hence, the net heat flow between the metering and guard boxes. Temperature of air in the guard box is regulated, so that the thermopile reading does not exceed 0.05 F, averaging less than 0.01 F. Thus, heat transferred through the walls of the metering box is negligible.

The test section through which metered heat passes equals the area of the metering box, 32 in. wide and 60 in. long (13.33 sq ft). It includes two 16-in. framing sections, 5 ft long. The remainder of the panel, designated guard section, has one framing space on each side of the test section and 16 in. at the end of each of two center framing sections. Air is circulated past the test panel surface within the metering box at approximately 40 fpm in a direction that tends to aid natural convection. Metering box air temperature can be regulated between 60 and 160 F with a differential of 0.05 F.

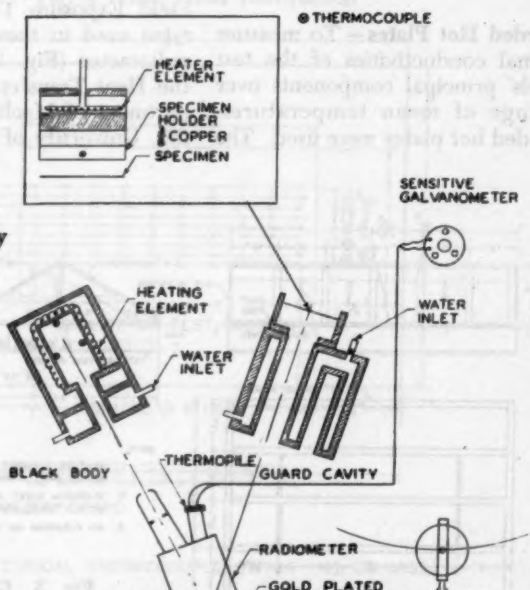


Fig. 2 Emissivity apparatus

Heat input to the metering box is measured by a watthour meter readable to 0.1 watthour. It measures total electrical energy consumed in the metering box, which includes energy dissipated by heating elements, auto transformers, fan motor and a small relay.

Tests were conducted in a 24-hr period, readings taken at half hr intervals. The final results are based upon uniform heat input and constant panel temperatures for a duration of a least 8 hr. Thermocouple voltages were measured with a manually operated precision potentiometer. Tests on the same panel on succeeding days without shutdown showed agreement to within 0.5%.

Guarded Hot Plates – To measure thermal conductivities of the test panels' principal components over a range of mean temperatures, guarded hot plates were used. The apparatus complies with ASTM Designation C177-45: "Thermal Conductivity of Materials by Means of the Guarded Hot Plate." These were two different hot plates: a 12-in. apparatus for specimens less than 1.5 in. thick and a 24-in. apparatus for specimens up to 4 in. thick. Tests could be conducted at any mean temperature between zero and 200 F, with the temperature difference across the test specimen being 50 F.

Emissivity Determination Apparatus – Emissivity determinations have been made on the different membranes used in the test program and on specimens from the Field Exposure Tests. The apparatus used in these studies was a radiometer (Fig. 2) developed by the Heat Transfer Div of the Department of Mechanical Engineering, University of Minnesota.

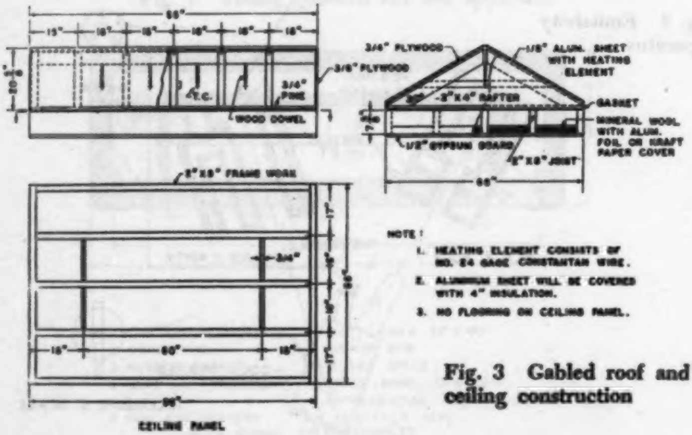


Fig. 3 Gabled roof and ceiling construction

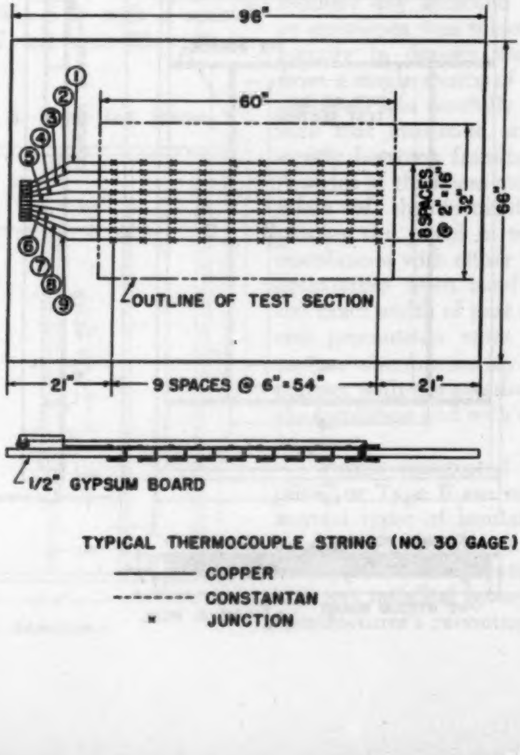
Attic Ceiling Test Apparatus — A pitched-roof-ceiling combination with a special heat meter was constructed to simulate an attic ceiling with no flooring (Fig. 3). Tests were conducted for both summer and winter conditions.

Summer tests (heat flow down) were made in the constant temperature room. The heat flow was induced by electrically heated aluminum plates attached to the outside surface of the roof. The constant temperature room was the heat sink or cold side environment of the test apparatus. Above the roof (not shown in Fig. 3), on the gable and periphery of the ceiling panel, 3-in. insulation was placed so the bulk of heat would pass down through the ceiling. Actual heat flow rate through the ceiling was measured by a heat meter incorporated into the interior finish of the panel.

For winter tests (heat flow up), aluminum heating plates and guard insulation were removed, and the apparatus was installed on the guarded hot box. The heat meter was omitted.

Heat Meter—Fig. 4 shows the heat meter, which is a ½-in. gypsum

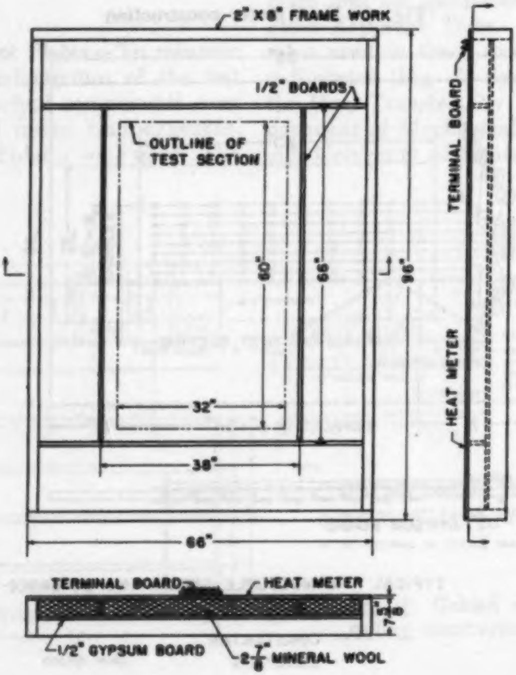
Fig. 4 Heat meter construction



board containing nine uniformly spaced rows of ten 30-gauge copper-constantan differential thermocouple junctions connected in series. The effective portion of the heat meter located concentrically with the test panel is 16 x 54 in. Thus, the heat meter is effective from the center of one joist space to the next, over a distance approximately equal to that of the test section in the guarded hot box (Fig. 5).

A special panel shown in Fig. 5 was used for calibrating the heat meter in the guarded hot box. This consists of 2⅞-in. mineral wool, the heat meter adjacent to the hot box and ½-in gypsum board on the cold side. The panel contains no joists or air spaces, so heat flux is essentially uniform. The heat meter was calibrated at 30, 40 and 50 F on the cold side and at 80 F on the warm side. Additional tests were made to evaluate the effect of mean temperature on heat meter calibration. These results showed that calibration is a linear function of temperature. Readings throughout the test program were thus corrected for the mean temperature of

Fig. 5 Heat meter calibration panel

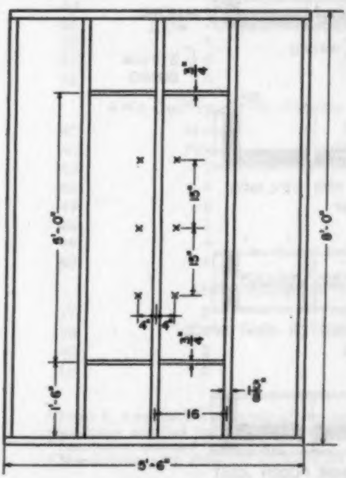


the meter. Interior finish in all of the tests was provided simultaneously by the heat meter for heat flow down and the majority of tests for heat flow horizontally and heat flow up. Heat flow rate, therefore, was obtained by both the hot box and heat meters.

DESCRIPTION OF TEST PANELS

The basic test frame (Fig. 6) consists of 2 x 8-in. joists, 16 in. on centers for the ceilings. Walls are 2 x 4-in. studs, 16 in. on centers. Interior finish is ½-in. gypsum board, and the exterior ¾-in. plywood, the latter representing flooring when used in ceiling panels or the exterior finish for wall panels. All frames have an outside dimension of 5.5 x 8 ft with a special blocked-off test area 32 x 60 in.

Fig. 6 Basic test frame



Joists or studs constitute the exterior of the test area in the 60-in. direction with ¾-in. soft pine headers between framing at each end. Isolation of the test area is necessary to prevent circulation of air between the test and outer guard areas. Periphery of the outer guard area (Fig. 7) is further insulated with 2-in. insulation, a precautionary measure to reduce the effect of radiant heat transfer from the periphery of the test area.

Two methods were used for applying insulation. "Laboratory Applied," designated Type L, is principally used throughout these studies to eliminate as many variables which are normally found in "Field Applications." Mineral wool without any attached membranes or envelopes was selected for uniformity in density and thickness from a single source of supply. Installation was carefully made to insure that insulation was installed snugly between framing members. A series of thickness readings were taken of the installation before placing the panel in test. Surface membranes with either high or low emissivities were carefully cut to the exact width of joist space. Special precautions were taken that surface membranes lay flat and in contact with the exterior surface of the insulation and with edges taped in place.

Panels designated "Field Applied" or Type F are regular commercial types of insulation, enveloped with reflective or non-reflective types of membranes. Insulation was installed according to the manufacturer's recommendations,

except header ends of membranes were sealed to prevent air leakage around ends of insulation. Because similar sealing materials were used, sealing did not alter the emissivity of the enclosing membrane.

Temperature measurements were taken with 30-gauge copper-constantan thermocouples. Surface temperatures, used to compute the conductance value, are an average obtained from readings at six different points on warm and cold surfaces of the panel. Locations of these points are shown in Fig. 6. Additional temperature measurements were taken at each interface in the panel midway between framing members at the midsection of the panel.

Five types of panels, combined with different thicknesses and types of insulation having reflective and non-reflective type membranes were used (see Fig. 7).

A. 2 x 8-in. ceiling joists with flooring.

B. 2 x 4-in. wall studs.

C. 2 x 8-in. ceiling joists without flooring.

D. 2 x 8-in. ceiling joists with reflective insulation only within the center of the joist space. Three different types were included.

E. 2 x 8-in. ceiling joists with insulation enclosed in a membrane and flanged to the sides of joists.

Fig. 7 Types of test panels

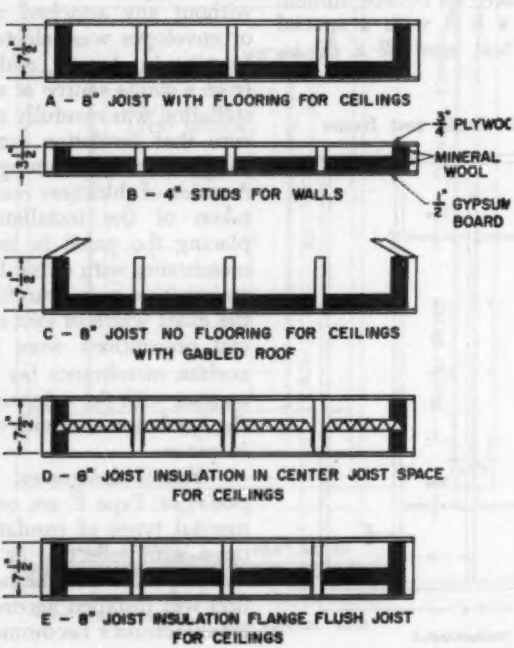


TABLE I
DESCRIPTION OF PANELS WITH TYPE L APPLICATION[1]

Panel No.	Thickness-In. Insulation	Air Space	Type Of Membrane	Reference Figure No.
Walls — 2 x 4-in. Studs — Heat Flow Horizontal				
30	None	3.625	Foil	7B
31	2	1.625	Kraft	7B
32	3	0.625	Kraft	7B
33	2	1.625	Foil	7B
34	3	0.625	Foil	7B
Ceilings — With Flooring — 2 x 8-in. Joists — Heat Flow Up				
24	None	7.5	None	7A
25	None	7.5	Foil	7A
26	2"	5.5	Kraft	7A
27	4"	3.5	Kraft	7A
28	2"	5.5	Foil	7A
29	4"	3.5	Foil	7A
Ceilings — with Flooring — 2 x 8-in. Joists — Heat Flow Down				
1	None	7.5	None	7A
2	2	5.5	None	7A
3	2	5.5	Foil	7A
5	2	5.5	Kraft	7A
8	4	3.5	Kraft	7A
9	4	3.5	Foil	7A
12	6	1.5	Kraft	7A
13	6	1.5	Foil	7A
19	None	7.5	Foil	7A
51	None	7.5	Special[3]	7A
52	None	7.5	"	7A
53	None	7.5	"	7A
54	2	5.5	"	7A
55	2	5.5	"	7A
56	4	3.5	"	7A
57	4	3.5	"	7A
58	4	3.5	"	7A
Attic Ceilings — No Flooring — 2 x 8-in. Joists — Heat Flow Down				
43	None	See	None	7C
44	None	Note[2]	Foil	7C
45	2	"	Kraft	7C
46	4	"	Kraft	7C
47	6	"	Kraft	7C
48	2	"	Foil	7C
49	4	"	Foil	7C
50	6	"	Foil	7C
Attic Ceilings — No Flooring — Heat Flow Up				
37	2	See	Kraft	7C
38	4	Note[2]	Kraft	7C
40	2	"	Foil	7C
41	4	"	Foil	7C

[1] Type L Application consists of insulation and membrane applied individually for uniformity.
[2] Insulation exposed to attic air.
[3] Membranes with different emissivities.

The program consisted of the following combination of tests:

1. No insulation and 2, 4 and 6-in. thick mineral fiber insulation with reflective and non-reflective types of membranes, applied independently to the exterior surface of insulation facing the air space. These tests are designated "Laboratory Applied" or Type L.

2. "Field Applied" tests, designated Type F, 2 and 3-in. thick batt insulation enclosed in reflective and non-reflective membranes.

3. Three types of reflective insulation, X, Y, and Z, (Fig. 8) were installed in the center of the joist space. Type X consisted of two sheets of aluminum foil separated 1½ in. by triangular kraft paper and Type Y of two sheets of aluminum foil having a backing of kraft paper as the outside membrane. A third sheet of aluminum foil was placed in the center of the air space and separated by an air space ⅞ in. from the two outside sheets. Two air spaces were each faced with one kraft surface and one aluminum surface. Type Z had two outside surfaces of aluminum foil with a sheet of kraft paper midway between an air space of approximately 1½ in. separating the aluminum foil.

Details of panels included in this program have been listed in Tables I and II.

RESULTS OF THERMAL CONDUCTIVITY TESTS ON PANEL COMPONENTS

Thermal conductivity tests were conducted upon each component used in the test panels, except the

TABLE II
DESCRIPTION OF PANELS WITH FLOORING ON 2 x 8-IN. JOISTS
TYPE F APPLICATION[1]

Panel No.	Insulation and Type of Application	Cold Side	Warm Side	Reference Figure No.
	Heat Flow Down			
101	2 in. Kraft enclosed, flanged on Joist edge	No	Yes	7A
102	2 in. Foil " " " " "	No	Yes	7A
103	3 in. Kraft " " " " "	No	Yes	7A
104	3 in. Foil " " " " "	No	Yes	7A
105	Type X Reflective, center of joist space	Yes	Yes	7D, 8
106	Type Z " " " " "	Yes	Yes	7D, 8
107	2 in. Kraft enclosed, flanged in joist space	Yes	Yes	7E
108	2 in. Foil " " " " "	Yes	Yes	7E
109	3 in. Kraft " " " " "	Yes	Yes	7E
110	3 in. Foil " " " " "	Yes	Yes	7E
111	Type Z Reflective, flanged on joist edge	No	Yes	7A, 8
112	Type Y* Reflective, center of joist space	Yes	Yes	7D, 8
113	Type Z Reflective, center of joist space	Yes	Yes	7D, 8
	Heat Flow Up			
114	Type Y* Reflective, center of joist space	Yes	Yes	7D, 8
115	Type X Reflective, center of joist space	Yes	Yes	7D, 8
116	Type Z Reflective, center of joist space	Yes	Yes	8D, 8

[1] Type F application consists of insulation "as received".
* Special precautions taken to obtain a good installation in this panel.

TABLE III
AVERAGE THERMAL CONDUCTIVITIES OF PANEL COMPONENTS AT DIFFERENT MEAN TEMPERATURES

Material	Mean Temperature, F. 75	100	125	150
Plywood	0.746	0.761	0.776	0.791
Gypsum Board	1.40	1.40	—	—
Mineral Wool Insulation	0.267	0.288	0.309	0.330

framing members. These were assumed to have conductivity K= 0.89, as recommended by the ASHRAE GUIDE. Thermal conductivity coefficients of representative samples of the ½-in. gypsum board, the ¾-in. plywood, and mineral wool insulation were determined over a range of mean temperatures by the Guarded Hot Plate method. Results of these tests are shown in Table III.

An additional series of thermal conductivity determinations was conducted upon the mineral wool insulation to establish the variation in the thermal conductivity coefficients which may be anticipated between the different lots which were received. Three different mean temperatures were used. Before tests on mineral wool enclosed batts, kraft paper or reflective coverings were removed. Results are in Table IV. Maximum variation in thermal conductivity coefficients of the mineral wool insulation is only 5.3% for mean temperatures of 75 and 100 F and 5.9% for 125 M.T. Considering the insulation was received at intervals over a period of approximately two years, thermal conductivity variations are well within the tolerances which may be expected of commercial insulations.

TABLE IV
THERMAL CONDUCTIVITIES OF DIFFERENT MINERAL WOOL SPECIMENS AT VARIOUS MEAN TEMPERATURES

Material	Type[1]	Density	Thermal conductivity 75	100	Btu/hr-ft²-F/in. 125
2 in. Mineral Wool	a	3.06	0.266[2]	0.287	0.309
2 in. Mineral Wool	b	2.98	0.263	0.284	0.305
3 in. Mineral Wool	a	3.17	0.262	0.282	0.302
4 in. Mineral Wool	b	2.96	0.266	0.289	0.311
2 in. Kraft enclosed Mineral Wool	b	2.97	0.266	0.287	0.308
3 in. Kraft enclosed Mineral Wool	b	3.35	0.264	0.285	0.306
2 in. Foil enclosed Mineral Wool	b	2.81	0.272	0.294	0.317
3 in. Foil enclosed Mineral Wool	b	3.54	0.258	0.279	0.299
Average		3.11	0.265	0.286	0.307
Max. Variation %		+14	+2.6	+2.8	+3.3
Min. Variation %		−9.6	−2.7	−2.5	−2.6

[1] a—Material received prior to July 1, 1957.
b—Material received after July 1, 1957
[2] Average of 11 tests.

RESULTS OF EMISSIVITY TESTS

Emissivity tests of kraft paper or aluminum foil membranes used in the panels for the Guarded Hot Box Tests produced the following results:

Type	Membrane	Temp. F.	Emissivity Range	Avg. Emissivity
L	Kraft	201	.869—.873	.871
L	Perforated foil on kraft paper	204	.029—.033	.032
F	Kraft	205		.877
F	Perforated foil	210		.023

Perforated aluminum foil enclosing insulation for the Type F application has an average emissivity of 0.023. Precautions were exercised in handling reflective materials to avoid changing surface emissivity. Further tests on materials exposed to normal handling showed no significant change in emissivity of surfaces.

RESULTS OF TESTS

Results of thermal conductance tests on panels described in Tables I and II are summarized in Tables V through IX. Winter conditions were conducted with outside temperature 20 F for "horizontal heat flow through walls," "heat flow up through ceilings" and "attic combinations with and without attic flooring." For summer conditions the outside air temperature was maintained at 160 F for "horizontal heat flow through the walls." "Heat flow down through floored ceilings" had air adjacent to the flooring maintained at 130 and 160 F for

TABLE V
THERMAL TRANSMITTANCE THROUGH WALLS FOR SUMMER AND WINTER CONDITIONS[1] HEAT FLOW HORIZONTAL

Panel No.	Insulation In.	Panel Temp F Mean	Panel Temp F Diff.	Air Space Mean	Air Space Diff.	Test C	Test U*	Guide U'.	Guide U'/U
				Reflective Membranes — Type L Application					
30	0	118.4	66.7	117.4	34.3	0.267	0.213	0.212	1.00
30	0	49.4	41.3	49.6	21.3	0.230	0.193	0.209	1.08
33	2	118.2	76.7	140.4	18.6	0.109	0.099	0.088	0.89
33	2	48.6	48.5	33.9	11.6	0.093	0.086	0.088	1.02
34	3	118.3	77.6	146.7	9.0	0.097	0.089	0.072	0.81
34	3	48.6	49.6	29.5	4.3	0.081	0.076	0.072	0.95
				Kraft Membranes — Type L Application					
31	2	118.4	74.9	144.7	5.7	0.130	0.115	0.102	0.89
31	2	48.9	37.6	31.6	4.4	0.104	0.095	0.102	1.07
32	3	118.4	76.9	148.6	3.5	0.103	0.094	0.080	0.85
32	3	48.6	48.9	29.2	3.0	0.084	0.078	0.080	1.03

* U Values based upon $f_i = 1.46$, $f_o = 4.00$ for summer; $f_i = 1.46$, $f_o = 6.00$ for winter.
[1] Summer conditions: $t_i = 75$ F, $t_o = 160$ F.
Winter conditions: $t_i = 75$ F, $t_o = 20$ F.

each of the panels. A limited number of tests were conducted at 100 F. These were discontinued in order to reduce the number of tests, and since the results added no significant information not obtained at the two higher temperatures, they are not included. For ceiling and attic combinations, outside surfaces of the roof were maintained at 130 and 160 F to simulate summer conditions with exposure to solar radiation. Inside air temperatures were maintained at 75 F in all tests.

Six individual thermocouple readings were recorded for each of four temperature conditions: inside air temperature, outside air temperature, surface temperature of the ½-in. gypsum board, and surface temperature of the ¾-in. plywood. Averages of each group were used in the final results. Locations of surface thermocouples are shown in Fig. 6. They are the surface temperature readings used in calculating conductances of panels determined by tests.

The conductance designated "C" in the Tables is determined as follows:

$$C = \frac{H}{\Delta ts} \tag{3}$$

where: C = panel conductance by test, Btu/hr-ft²-F
H = measured heat flow rate, Btu/hr-ft²
Δts = surface temperature difference F

Transmittance coefficient "U" is determined by adding respective surface coefficients, recommended

TABLE VI
THERMAL TRANSMITTANCE THROUGH CEILINGS WITH FLOORING — HEAT FLOW UP[1]

Panel No.	Insulation In.	Panel Temp F Mean	Panel Temp F Diff.	Air Space[2] Mean	Air Space[2] Diff.	Test C	Test U*	Guide U'	Guide U'/U
			Reflective Membranes — Type L Application						
25	0	49.2	40.9	49.4	22.6	0.269	0.203	0.201	0.99
28	2	48.4	48.9	30.8	11.4	0.093	0.083	0.082	0.99
29	4	48.2	50.9	28.8	7.2	0.058	0.054	0.055	1.02
			Kraft Membranes — Type L Application						
24	0	48.6	35.2	52.4	15.3	0.431	0.282	0.264	0.94
26	2	48.4	48.4	31.4	5.3	0.103	0.090	0.091	1.01
27	4	48.4	50.5	26.8	2.7	0.062	0.058	0.058	1.00
			Reflective Insulations — Type F Application						
114	Y	48.3	47.6	65.3	10.3	0.111	0.092	0.085	0.92
				33.8	8.9				
115	X	47.8	45.8	63.9	11.8	0.136	0.108	0.085	0.72
				35.6	11.9				
116	X	48.1	47.5	63.9	11.2	0.116	0.095	0.085	0.89
				34.4	8.8				

* U values based upon $f_i = f_o = 1.63$ for heat flow up.
[1] Winter conditions: t_i = 75 F, t_o = 20 F.
[2] Where two air space mean temperatures are shown, the insulation was flanged between joists.

TABLE VII
THERMAL TRANSMITTANCE THROUGH CEILINGS WITH FLOORING — HEAT FLOW DOWN[1]

Panel No.	Insulation In.	Panel Temp F Mean	Panel Temp F Diff.	Air Space[2] Mean	Air Space[2] Diff.	Test C	Test U*	Guide U'	Guide U'/U
Reflective Membranes — Type L Application									
19	0	103.8	45.1	102.1	36.1	0.138	0.110	0.086	0.77
19	0	120.0	68.9	117.3	54.1	0.146	0.115	0.086	0.74
3	2	103.2	49.1	112.7	22.6	0.081	0.071	0.050	0.75
3	2	118.6	76.1	133.8	33.7	0.085	0.073	0.050	0.73
9	4	103.5	50.0	119.2	13.8	0.055	0.050	0.043	0.86
9	4	118.6	78.3	143.5	20.3	0.058	0.053	0.043	0.76
13	6	103.0	51.6	122.0	9.5	0.045	0.042	0.038	0.90
13	6	118.4	79.5	149.7	13.6	0.047	0.043	0.038	0.88
Reflective Membranes — Type F Application									
102	2	103.4	48.5	114.7	18.1	0.087	0.075	0.056	0.75
102	2	119.0	74.9	136.9	26.3	0.091	0.078	0.056	0.72
108	2**	103.0	50.3	118.0	14.2	0.071	0.063	0.046	0.73
				84.1	10.4				
104	3	103.4	49.1	117.4	14.3	0.075	0.066	0.048	0.73
104	3	118.8	76.0	141.0	20.9	0.079	0.067	0.048	0.72
110	3**	103.0	50.6	119.3	13.4	0.069	0.061	0.041	0.67
				82.7	8.2				
105	X	103.6	48.6	116.5	16.0	0.082	0.071	0.046	0.65
				87.6	14.8				
105	X	119.0	75.0	139.7	23.0	0.088	0.076	0.046	0.61
				94.9	23.5				
106	Z	103.0	49.2	118.7	11.9	0.073	0.064	0.044	0.69
				88.7	17.4				
106	Z	118.9	76.2	142.7	17.3	0.073	0.068	0.044	0.65
				96.2	27.6				
111	Z**	103.4	47.0	112.6	18.0	0.128	0.103	0.044	0.43
112	Y**	103.2	50.1	117.7	15.2	0.072	0.064	0.046	0.72
				86.7	15.3				
113	Y**	103.0	50.3	117.9	14.6	0.070	0.062	0.046	0.74
				87.1	16.7				
Kraft Membranes — Type L Application									
1	0	105.3	34.6	100.5	16.1	0.411	0.234	0.222	0.95
1	0	122.4	53.2	115.8	23.2	0.428	0.239	0.222	0.93
2, 5	2	100.6	40.9	119.1	4.6	0.125	0.101	0.085	0.84
2, 5	2	119.3	72.8	143.3	6.5	0.126	0.102	0.085	0.83
8	4	103.3	49.6	123.5	2.7	0.072	0.064	0.055	0.86
8	4	118.8	76.7	150.0	3.9	0.075	0.066	0.055	0.83
12	6	103.1	51.0	125.3	2.0	0.052	0.047	0.042	0.89
12	6	118.6	78.8	152.8	2.8	0.054	0.049	0.042	0.86
Kraft Membranes — Type F Application									
101	2	103.2	47.7	119.8	4.4	0.115	0.095	0.085	0.90
101	2	119.4	72.5	144.8	6.2	0.117	0.096	0.085	0.89
107	2**	103.3	48.2	121.0	3.9	0.099	0.084	0.080	0.95
				82.4	4.5				
103	3	103.4	48.4	122.0	3.2	0.092	0.078	0.066	0.84
103	3	118.2	74.5	147.9	4.6	0.094	0.080	0.066	0.83
109	3**	103.2	49.5	122.9	2.9	0.080	0.069	0.065	0.94
				81.3	3.1				

* U Value based upon $f_i = f_o = 1.08$ U' based upon ASHAE GUIDE Calculations.

[1] Summer conditions: $t_i = 75$ F $t_o = 130$ and 160 F, except those designated with ** which were conducted at $t_o = 130$ F only.

[2] Where two air space mean temperatures are shown, the insulation was flanged between joists.

TABLE VIII
THERMAL TRANSMITTANCE THROUGH CEILINGS WITHOUT FLOORING AND WITH ATTIC — HEAT FLOW DOWN

Panel No.	Insulation In.	Panel Temp F Mean	Panel Temp F Diff.	Air Temp F Inside	Air Temp F Attic	Test C	Test U*	Guide U'	Guide U'/U
Reflective Membranes — Type L Application									
44	0	82.3	2.2	75	118.4	2.860	0.172	0.161	0.94
44	0	86.4	3.6	75	141.7	2.880	0.172	0.161	0.94
48	2	90.9	24.8	75	121.9	0.160	0.085	0.076	0.89
48	2	100.4	39.3	75	146.9	0.167	0.087	0.076	0.87
49	4	94.4	33.6	75	123.2	0.084	0.058	0.052	0.90
49	4	105.6	52.8	75	149.2	0.088	0.059	0.052	0.88
50	6	95.8	37.5	75	132.9	0.056	0.043	0.041	0.95
50	6	107.9	58.6	75	150.4	0.059	0.045	0.041	0.91
Kraft Membranes — Type L Application									
43	0	92.9	5.8	75	110.3	2.710	0.450	0.406	0.90
45	2	98.8	37.6	75	120.3	0.150	0.118	0.101	0.86
45	2	112.4	58.1	75	145.3	0.154	0.120	0.101	0.84
46	4	99.8	43.6	75	122.5	0.079	0.069	0.062	0.90
46	4	113.6	66.9	75	148.2	0.084	0.073	0.062	0.85
47	6	100.3	46.2	75	123.5	0.054	0.049	0.046	0.94
47	6	114.4	71.0	75	150.1	0.058	0.051	0.046	0.89

* U values based upon $f_i = 1.08$, $f_o = 1.08$ for Kraft Membranes and $f_o = 0.22$ for reflective membranes. Roof surface temperatures = 130 and 160 F respectively.

for different conditions of heat flow by the ASHRAE GUIDE, to the test conductance "C."

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{C} + \frac{1}{f_o}} \quad (4)$$

Following is a summary of the surface coefficients used:

Two outside surface temperatures of the panel, measured at the surfaces of the ½-in. gypsum board and the ¾-in. plywood, were used to determine the panel mean temperatures and temperature differences. For "ceiling-attic combination without flooring," the outside surface temperatures were taken at the top surface of the insulation,

1. Heat flow horizontal through walls:
   - Summer $f_i = 1.46$, $f_o = 4.00$
   - Winter $f_i = 1.46$, $f_o = 6.00$
2. Heat flow through ceilings with flooring:
   - Summer $f_i = 1.08$, $f_o = 1.08$
   - Winter $f_i = 1.63$, $f_o = 1.63$
3. Heat flow through ceilings without flooring:
   - Summer $f_i = 1.08$, $f_o = 1.08$ for kraft
   - $f_i = 1.08$, $f_o = 0.22$ for reflective
   - Winter $f_i = 1.63$, $f_o = 1.63$ for kraft
   - $f_i = 1.63$, $f_o = 0.76$ for reflective

or the ½-in. gypsum board, when insulation was omitted.

Air space mean temperatures and temperature differences are based upon temperature measurements at each interface in the panel midway between the framing members at the midsection of the metered area.

METHOD FOR CALCULATING COEFFICIENTS U′

Transmittance value identified by symbol U′ in the summary tables is a calculated coefficient using procedures recommended in Chapter 9 of the 1960 ASHRAE GUIDE. Since most engineers and architects are primarily dependent upon the ASHRAE GUIDE for information relating to heat transfer through structures, selection of conductance and conductivity coefficients for different panel components are those recommended in GUIDE Table 4, for building and insulating materials. Except for the ½-in. gypsum board, GUIDE coefficients closely agree with values obtained by tests:

		GUIDE	Test
½-in. Gypsum board	"C"	2.25	2.80
¾-in. Plywood	"C"	1.07	1.02
Mineral Wool	"k"	0.27	0.265 M.T. = 75
		0.27	0.286 M.T. = 100
Framing members	"k"	0.80	—

The ultimate effect of the gypsum board's lower resistance and the slightly higher resistance of the ¾-in. plywood is negligible within the range of this test transmittance values. The difference in combined resistances of the ½-in. gypsum board and ¾-in. plywood, based upon the GUIDE and actual tests on these two materials, is 1.388 − 1.339 or 0.049. Calculated U′ coefficients' values would be increased only 0.25% to 0.5%, if actual test values were used in place of the GUIDE value for a range of transmittance coefficients 0.05 to 0.10, respectively.

Based upon tests, average emissivity values of surface membranes facing air spaces are 0.87 for kraft and 0.032 for reflective materials. Effective emissivities given in the GUIDE (0.82 for non-reflective surfaces and 0.05 for reflective surfaces) were used in determining the calculated U′ coefficients.

To determine calculated transmittance coefficients for the different directions of heat flow, the following procedures were used:

1. Symbols:
   - U = Transmittance coefficient by test
   - U′ = Transmittance coefficient by GUIDE calculations
   - U_i = Transmittance coefficient for area between framing members
   - U_s = Transmittance coefficient backed by framing members
   - S = Percentage of area backed by framing members
   - E = Effective emissivity of air spaces.

2. Heat flow up and down: Transmittance coefficients of uninsulated panels between framing members is determined from GUIDE Tables 3 and 4, and, as recommended by the GUIDE, no correction is made for changes in air space temperature differences

for heat flow down. Where the mean temperatures of air spaces are substantially different, for those listed for heat flow down in the Tables resistance of the air space was extrapolated from resistance of air space versus mean temperatures, given in GUIDE Table 3-C. For heat flow up, resistances of the air space were extrapolated for the temperature differences. Mean temperatures in this particular case are similar to test conditions, so no correction was considered. U_t was determined from GUIDE Table 16, parts C and D. The U′ or average U value is the corrected value for framing members using GUIDE Fig. 6.

3. Horizontal Heat Flow:
Essentially the same procedure is followed. Air space conductance was corrected for temperature differences whenever necessary by extrapolating the values in GUIDE Table 3-C. No correction is introduced for temperature differences in test and GUIDE mean temperatures since this effect is minor. However, 50 and 90 F mean temperatures were used for winter and summer conditions, respectively.
In some instances, the range of values in GUIDE Table 16 was inadequate to complete calculations; for example, panels with special reflective types of insulation and mineral wool insulated panels with extremely low transmittance values. U_t in these cases was calculated from GUIDE Tables 3 and 4.

COMPARISON OF TEST AND CALCULATED TRANSMITTANCE COEFFICIENTS

Tables V through IX present a summary of test results, comparing test and calculated transmittance values for different directions of heat flow and air space mean temperatures. Tables V through IX show the ratios of U′/U as approximately the same for outside air temperatures of 130 and 160 F, for the same type panel. In Table X, these two test conditions have been combined, and average mean temperatures as well as average U′/U for a specific panel are presented. Discussion in this section applies only to comparison of "U" values by test and by calculation. It does not include evaluation of kraft or foil membranes as components of the over-all construction.

TABLE IX
THERMAL TRANSMITTANCE THROUGH CEILINGS WITHOUT FLOORING AND WITH ATTIC — HEAT FLOW UP

Panel No.	Insulation In.	Panel Temp F Mean	Panel Temp F Diff.	Air Temp F Inside	Air Temp F Attic	Test C	Test U*	Guide U′	Guide U′/U
			Reflective Membranes — Type L Application						
40	2	54.8	35.7	75	31.3	0.132	0.106	0.100	0.95
41	4	52.8	41.6	75	27.9	0.072	0.063	0.061	0.97
			Kraft Membranes — Type L Application						
37	2	53.2	38.7	75	31.2	0.130	0.112	0.108	0.96
38	4	51.0	45.0	75	27.3	0.071	0.065	0.064	0.98

* U value based upon $f_i = 1.63$, $f_o = 1.63$ for Kraft Membrane and $f_o = 0.76$ for reflective membranes. Outside air —20 F.

TABLE X
COMPARISON OF TRANSMITTANCE COEFFICIENTS BY TEST AND BY CALCULATION

Insulation In.	Type Application	Condition	Type of Surfaces	Heat Flow Direction	Air Space M.T. F	U'/U
			Wall Panels			
0	L	Summer	Foil	Horizontal	117.4	1.00
2	L	Summer	Foil	Horizontal	140.4	0.89
3	L	Summer	Foil	Horizontal	146.7	0.81
2	L	Summer	Kraft	Horizontal	144.7	0.89
3	L	Summer	Kraft	Horizontal	148.6	0.85
0	L	Winter	Foil	Horizontal	49.6	1.08
2	L	Winter	Foil	Horizontal	33.9	1.02
3	L	Winter	Foil	Horizontal	29.5	0.95
2	L	Winter	Kraft	Horizontal	31.6	1.07
3	L	Winter	Kraft	Horizontal	29.2	1.03
			Ceiling Panels With Flooring			
0	L	Winter	Foil	Up	49.4	0.99
2	L	Winter	Foil	Up	30.8	0.99
4	L	Winter	Foil	Up	28.8	1.02
0	L	Winter	Kraft	Up	52.4	0.94
2	L	Winter	Kraft	Up	31.4	1.01
4	L	Winter	Kraft	Up	26.8	1.00
X*	F	Winter	Foil	Up	35.6	0.92
Y*	F	Winter	Foil	Up	33.8	0.72
Z*	F	Winter	Foil	Up	34.4	0.89
			Ceiling Panels With Flooring			
0	L	Summer	Foil	Down	109.7	0.75
2	L	Summer	Foil	Down	118.2	0.74
2	F	Summer	Foil	Down	120.8	0.73
2	F**	Summer	Foil	Down	118.0	0.73
3	F	Summer	Foil	Down	129.2	0.73
3	F**	Summer	Foil	Down	119.3	0.67
4	L	Summer	Foil	Down	131.3	0.81
6	L	Summer	Foil	Down	135.8	0.89
0	L	Summer	Kraft	Down	108.2	0.94
2	L	Summer	Kraft	Down	131.2	0.84
2	F	Summer	Kraft	Down	132.4	0.90
2	F**	Summer	Kraft	Down	121.0	0.95
3	F	Summer	Kraft	Down	135.0	0.84
3	F**	Summer	Kraft	Down	122.9	0.94
4	L	Summer	Kraft	Down	136.8	0.84
6	L	Summer	Kraft	Down	139.0	0.88
X*	F	Summer	Foil	Down	128.1	0.63
Y*	F	Summer	Foil	Down	117.8	0.73
Z*	F	Summer	Foil	Down	130.7	0.67
Z*	F[1]	Summer	Foil	Down	112.6	0.43

* Special reflective types of insulation — no mineral wool.
** Mineral wool insulation flanged in center of joist spaces.
[1] Special reflective type insulation flanged on bottom of joists.

Ceiling-Attic Combinations—No Flooring

0	L	Summer	Foil	Down	130.0	0.94
2	L	Summer	Foil	Down	135.4	0.88
4	L	Summer	Foil	Down	136.4	0.89
6	L	Summer	Foil	Down	141.7	0.93
0	L	Summer	Kraft	Down	110.3	0.90
2	L	Summer	Kraft	Down	132.8	0.85
4	L	Summer	Kraft	Down	135.3	0.88
6	L	Summer	Kraft	Down	136.8	0.92
2	L	Winter	Foil	Up	31.3	0.95
4	L	Winter	Foil	Up	27.9	0.97
2	L	Winter	Kraft	Up	31.2	0.96
4	L	Winter	Kraft	Up	27.3	0.98

In the 1960 ASHRAE GUIDE:

Chapter 9: A word of caution is given in the use of Tables. "In calculating U values, exemplary conditions of components and installations are assumed, i.e., that insulating materials are uniformly of the nominal thickness and conductivity, that air spaces are of uniform thickness and surface temperatures, that effects due to moisture are not involved, and that installation details are in accordance with design. Some evidences of departures of measured values from calculated values for certain insulated constructions are given in 'Building Materials and Structures Report BMS 151', National Bureau of Standards. In order to provide a reasonable factor of safety to account for departures of constructions from exemplary conditions, in part due to field construction requirements and practices, some may wish, before making corrections for framing (as indicated in Fig. 6), to increase moderately the calculated U values of the insulated walls, floors and ceiling sections obtained from Table 16. Where reflective air spaces are involved, increase of U values up to 10 percent for applications where heat flow is horizontal or upward, and up to 20 percent where heat flow is downward, appear reasonable on the basis of present information."

In this program, exemplary conditions of components and installations were present and the variable usually found in construction is not included.

Considering the number of factors which may cause differences in test values and GUIDE calculations, there is good agreement between the two for the following conditions: (Table numbers refer to results)

1. Horizontal heat flow through walls under winter conditions (Table V), the ratio of U'/U varies from 0.95 to 1.08 with no apparent differences between reflective or non-reflective surfaces in combination with mineral wool insulation. For insulated wall panels shown in Fig. 7, insulation was adjacent to the interior surface finish with either kraft or perforated aluminum foil membrane adjacent to the exterior surface of insulation.

2. Heat flow up through ceilings with flooring (Table VI), the ratio U'/U varies from 0.94 to 1.02, or an average of 0.99 for both reflective and non-reflective surfaces with and without mineral wool insulation. However, for special types of multiple layers of foil (Fig. 8), ratio variation is from 0.72 to 0.92.

3. Ceiling and attic combina-

tions without flooring with heat flow up (Table IX), the ratio of U′/U varies between 0.95 and 0.98 for insulation enclosed in either a reflective or non-reflective membrane. These results show good agreement between U and U′.

In all of the cited conditions, air space mean temperatures are in the lower range of 26.8 to 52.4. This is significant in regard to heat flow through studs or joists which are exposed to the air spaces. As will be shown in the following discussion, agreement between U′ and U is not as satisfactory or as consistent, when air space mean

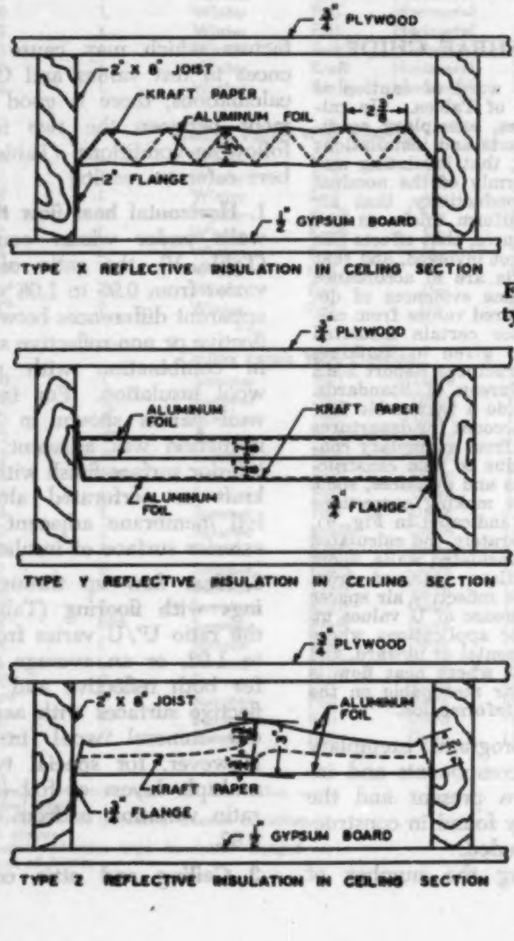


Fig. 8 Reflective type insulation

temperatures are 120 to 145 F under summer conditions.

1. Horizontal heat flow through walls under summer conditions (Table V), the ratio U'/U for the mineral wool insulation inclusive of reflective and non-reflective surfaces varies from 0.81 to 0.89. There is no apparent difference between the effect of kraft or foil surfaces. However, the ratio appears less for three-in. than for two-in. insulation. Air space mean temperatures vary from 140.4 to 148.6 F. For the non-insulated panel with foil adjacent to the ½-in. gypsum board, the ratio U'/U is 1.00 with the air space mean temperature equal to 117.4 F, or 25 to 30 F lower than insulated panels.

2. Ceiling-attic combination without flooring and heat flow down (Table VIII), the ratio of U'/U varies from 0.85 to 0.92 for the insulated panels with either kraft or foil membranes. Attic space mean temperature ranges from 110.3 to 141.7 F. There is no apparent difference between calculated and test values for either kraft or foil surfaces.

3. Ceiling panels with flooring and heat flow down (Table VII). Mineral wool insulated panels will be discussed separately from the special multiple layers of reflective insulation designated Types X, Y and Z. The ratios U'/U for mineral wool insulation of different thicknesses with kraft and foil surfaces are as follows: (Field type of application, F indicates face stapled and F*—inset stapled).

Insulation In.	Type of Application	Ratio U'/U Kraft	Ratio U'/U Foil
None		0.94	0.76
2	L	0.84	0.74
2	F	0.90	0.73
2	F*	0.95	0.73
3	F	0.84	0.73
3	F*	0.95	0.67
4	L	0.84	0.81
6	L	0.88	0.89

The ratio of U'/U varies from 0.67 to 0.75 for insulation with foil and thicknesses up to three in. When insulation thicknesses are 4 and 6 in. with foil, the ratio is 0.81 and 0.89, respectively.

Multiple reflective layers of foil, Types X, Y and Z (Fig. 8), have the ratio U'/U which varies from 0.43 to 0.73. 0.43 applies to Type Z reflective material face, stapled on the lower edges of joists.

In conclusion, further studies of reflective and non-reflective air space conductances for heat flow under summer conditions should be undertaken. This applies to both heat flow down and horizontal heat flow. Present methods of calculation of heat flow under winter conditions, either horizontal or upward, are satisfactory. Configuration of multiple layers of reflective insulations and variables, present in final application, indicate Guarded Hot Box Tests should be conducted on each type of insulation — X, Y and Z — under both summer and winter conditions. Present methods are inadequate for calculating overall transmittance coefficients for these types of insulation.

COMPARISON OF TRANSMITTANCE COEFFICIENTS WITH AND WITHOUT REFLECTIVE SURFACES

Only test conductances combined with respective surface conductances are used in this evaluation (Table XI). Test conditions for each type panel are identical, except reflective air spaces replace non-reflective air spaces. Test conductance values eliminate variables which may be inherent in calculated values. Symbols U_n and U_r are used to differentiate between non-reflective and reflective surfaces, respectively. The ratio U_r/U_n is used to evaluate effectiveness of reflective surfaces. These results are:

TABLE XI
HEAT TRANSFER THROUGH MINERAL WOOL WITH AND WITHOUT REFLECTIVE SURFACES UNDER DIFFERENT CONDITIONS OF HEAT FLOW

Panel No	Insulation In.	Type	Air Space M.T. Non-Refl	Refl	Condition	Test U* U_n	U_r	U_r/U_n
Wall Panels — Heat Flow Horizontal								
31-33	2	L	144.7	140.4	Summer	0.115	0.099	0.85
32-34	3	L	148.6	146.7	Summer	0.094	0.089	0.95
31-33	2	L	31.6	33.9	Winter	0.095	0.086	0.94
32-34	3	L	29.2	29.5	Winter	0.078	0.076	0.97
Ceiling Panels with Flooring — Heat Flow Up								
26-28	2	L	31.4	30.8	Winter	0.090	0.083	0.92
27-29	4	L	26.8	28.8	Winter	0.058	0.054	0.93
Ceiling Panels with Flooring — Heat Flow Down								
2, 5-3	2	L	119.1	112.7	Summer	0.101	0.071	0.70
2, 5-3	2	L	143.3	133.8	Summer	0.102	0.073	0.72
8-9	4	L	123.5	119.2	Summer	0.064	0.050	0.78
8-9	4	L	150.0	143.5	Summer	0.066	0.053	0.80
12-13	6	L	125.3	122.0	Summer	0.047	0.042	0.89
12-13	6	L	152.8	149.7	Summer	0.049	0.043	0.88
101-102	2	F	119.8	114.7	Summer	0.095	0.075	0.79
101-102	2	F	144.8	136.9	Summer	0.096	0.078	0.80
103-104	3	F	122.0	117.4	Summer	0.078	0.066	0.85
103-104	3	F	147.9	141.0	Summer	0.080	0.067	0.84
Ceiling Without Flooring and Attic Combinations — Heat Flow Down								
45-48	2	L	120.3	121.9	Summer	0.118	0.085	0.72
45-48	2	L	145.3	146.9	Summer	0.120	0.087	0.73
46-49	4	L	122.5	123.2	Summer	0.069	0.058	0.84
46-49	4	L	148.2	149.2	Summer	0.073	0.059	0.81
47-50	6	L	123.5	132.9	Summer	0.049	0.043	0.88
47-50	6	L	150.1	150.4	Summer	0.051	0.045	0.87
Ceiling Without Flooring and Attic Combinations — Heat Flow Up								
37-40	2	L	31.2	31.3	Winter	0.112	0.105	0.94
38-41	4	L	27.3	27.9	Winter	0.065	0.063	0.97

* U_n = Test Transmittance values for non-reflective surfaces. U_r = Test Transmittance values for reflective surfaces.

1. Horizontal Heat Flow Through Walls:

a) For summer conditions the ratio of U_r/U_n varies from 0.86 for 2-in. insulation to 0.95 for 3-in. insulation. Stud space is 3⅝ in., so air space is ⅝ in. for 3-in. insulation, compared with 1⅝ in. for 2-in. insulation.

b) For winter conditions, the ratio varies from 0.94 for 2-in. insulation to 0.97 for 3-in. Differences are in the same order found for summer conditions but of lower magnitude. However, air space mean temperatures are considerably lower for winter conditions.

2. Heat Flow Up Through Ceiling Panels with Flooring:

Mean temperatures of air space are in the low range of 26.8 to 31.8 F. The ratio of U_r/U_n is 0.92 and 0.93 for 2 and 4-in. insulation, respectively.

3. Heat Flow Down Through Ceiling Panels with Flooring:

Reduction in the heat transmittance coefficient with reflective air spaces varies inversely with the thickness of insulation:

	U_r/U_n
a) Type L, 2-in. insulation	0.71
b) Type L, 4-in. insulation	0.79
c) Type L, 6-in. insulation	0.89

4. Heat Flow Down Through Ceiling Without Flooring and with Gabled Roof:

The value of reflective surfaces is approximately the same for these conditions as for ceilings with flooring and Type L application. Average ratio of U_r/U_n is 0.72 for 2-in. insulation; 0.82 for 4-in., and 0.88 for 6-in.

5. Heat Flow Up Through Ceiling Without Flooring and with Gabled Roof:

Ratios of U_r/U_n are 0.92 and 0.95 for 2 and 4-in. insulation. These are in the same range as "walls with heat flow horizontal" and "ceilings with flooring and with heat flow up under winter conditions."

EFFECT OF ATTIC DUST ON EMISSIVITY OF ALUMINUM FOIL

An important consideration, regarding ceilings without flooring and with highly reflective surfaces facing the roof, is the effect of dust on emissivity of reflective surfaces and resultant increase in heat transfer. In order to determine the magnitude of this effect, a study was undertaken to supplement thermal transmittance tests. Results are presented in Table XII.

It was expected that dust accumulating over a long period of time would increase the emissivity value of reflective surfaces; the amount of increase would depend on attic ventilation, length of exposure time and the neighborhood of the home. These expectations were confirmed (Table XII). Emissivity values of specimens after one year of exposure range from .041 to .107. It is significant that the specimen taken from the unventilated attic showed least change in emissivity, whereas three specimens taken from ventilated attics showed a large increase. Of these three, two

homes located near open country showed considerably higher increase than the one located in a more developed area. It is interesting to note the similarity in the two highest values, since these homes are located in comparable neighborhoods and have similar ventilation conditions.

Emissivity results obtained after two years of exposure range from 0.143 to 0.545. The largest increase may be due to abnormal conditions, since the area was being developed and considerable grading was taking place. For the location showing the smallest increase in emissivity, values were lowest for each year. The magnitude of change, from 0.022 to 0.143 in two years, is notable, considering the home has the most favorable location and ventilation conditions for avoiding dust infiltration. The home located in open country with a vented attic had a large increase in emissivity from 0.022 to 0.361. Specimens were not available for location C because of absence of occupants.

Results after three years show no increase for location A. For location B, additional insulation was added to the ceiling during the third year of exposure, setting up an abnormal condition. Specimens were not protected, making this test meaningless because of improper handling and exposure to extraordinarily contaminated attic air conditions. The 1959 determinations for location C are based upon three specimens, the results: 0.409, 0.417 and 0.467. Lowest increase in emissivity during this period was obtained for location D.

EFFECT OF CHANGE IN SURFACE EMISSIVITY

A series of tests was conducted to determine the effectiveness of dif-

TABLE XII
EFFECT OF EXPOSURE TO ATTIC DUST ON EMISSIVITY OF ALUMINUM FOIL

Location	Residence Description	Attic Description	Emissivity Values August 1956	September 1957	July 1958	September 1959
A	Near open country — dusty 8	Vented at each gable, no fan, half floored	0.022	0.106	0.361*	0.341
B	Suburban Development	Ventilated attic with fan (fan rarely used), no flooring	0.022	0.067	0.545	—
C	Near open country — dusty	Vented under eaves, no flooring	0.022	0.107**	—	0.432
D	Built up section, no open country	Not ventilated, no flooring	0.022	0.041	0.143	0.207

* Attic was cleaned during the year
** Dried water spots observed on sample.

ferent surface emissivities upon heat flow through insulated and non-insulated ceilings. Surface membranes were composed of three materials: perforated aluminum foil (e = 0.032); reflective coated paper (e = 0.172) and kraft paper (e = 0.87). Two additional membranes were obtained by combining these materials and were constructed by taping narrow strips of higher emissivity materials onto a sheet of lower emissivity. Strips were equal in length to the width of joist space and placed at regular intervals perpendicular to joists. A combination of reflective and aluminum foil was the basis for e = 0.104, kraft and aluminum foil for e = 0.382.

A summary of these test results for panels with no insulation, 2-in. and 4-in. mineral wool is shown in Table XIII. Test conductances are used to obtain direct comparison. The ratio of conductances of C_e'/C_e, in the last column, is based upon conductances with kraft paper, e = 0.87, compared with the conductances of membranes with lower emissivity values. Effectiveness of reflective surfaces diminishes as the emissivity increases, especially in combination with mineral wool insulation. As emissivity of foil increases from 0.032 to 0.382, thermal conductances, when used in combination with mineral wool insulation, approach that of kraft paper or e = 0.87. The increase in emissivity reduces foil effectiveness to a greater degree as the insulation thickness is increased. For 2-in. insulation

TABLE XIII
EFFECT OF SURFACE EMISSIVITY ON HEAT FLOW DOWN THROUGH CEILINGS WITH FLOORING

Panel No	Air Space F Mean	Air Space F Diff	Surface Emissivity	Test Conductance	C'e/Ce[1]
			No Insulation		
1	100.5	16.1	0.870	0.411	1.00
53	102.1	21.5	0.382	0.296	0.72
52	102.1	28.2	0.172	0.227	0.55
51	102.1	30.8	0.104	0.197	0.48
19	102.0	35.8	0.032	0.168	0.41
			2-in. Insulation		
5	115.9	14.0	0.870	0.124	1.00
55	119.1	6.7	0.382	0.109	0.88
4	113.0	21.7	0.172	0.099	0.80
54	116.1	14.6	0.104	0.093	0.75
3	112.7	22.6	0.032	0.081	0.65
			4-in. Insulation		
8	123.4	3.0	0.870	0.072	1.00
58	122.9	4.7	0.382	0.070	0.97
57	126.9	7.4	0.172	0.067	0.93
56	120.9	9.33	0.104	0.063	0.87
9	119.2	13.8	0.032	0.055	0.77

[1] C'e = emissivity of respective surface below 0.87. Ce = emissivity of kraft paper 0.87.

this reduction is 65.5% and for 4-in. insulation, 87%. For non-insulated structures, the loss is more than 50% of its original effectiveness when emissivity increases from 0.032 to 0.382.

CONCLUSION

The GUIDE method of calculating thermal transmittance coefficients for walls and ceiling containing air spaces in combination with mineral wool insulation, with heat flow under winter conditions, agrees closely with test values, for either reflective or non-reflective surfaces.

Agreement is not as satisfactory for heat flow under summer conditions, and an even greater difference occurs for reflective surfaces than for non-reflective surfaces. Differences between the calculated and test values decrease as thickness of the mineral wool insulation increases. Further studies should be instituted with heat flow down on air space conductances enclosed in conventional joist spaces which are 4 in. or more deep and insulated with high and low emissivity materials.

With the present available information, it is difficult to calculate with satisfactory accuracy transmittance coefficients for special types of reflective insulation having multiple layers of foil. Guarded Hot Box Tests should be conducted on the material installed in place as recommended by manufacturers.

Further studies are needed for the effect of surface contamination on reflective surfaces exposed to field conditions, and the effect upon overall heat transfer. These limited studies indicate that this may be a serious factor after exposure of only a few years.

ACKNOWLEDGMENTS

The authors wish to express their appreciation to the National Mineral Wool Association and their Technical Committee for financial assistance in making these studies possible. The NMWA Technical Committee also has contributed consistently many valuable suggestions throughout the period of the program. Our appreciation is also extended to Professor A. B. Algren, University of Minnesota, for his continuous interest and valuable suggestions during the execution of this program. We also are indebted to Merle L. Erickson, formerly Research Associate in the Department of Mechanical Engineering, University of Minnesota, for his valuable technical assistance in preparing this paper.

REFERENCES

"Thermal Resistance of Airspaces and Fibrous Insulation Bounded by Reflective Surfaces" United States Department of Commerce, National Bureau of Standards, B.M.S. Report 151, November 14, 1957.

ASHRAE GUIDE, 1959.

"The Thermal Insulating Value of Airspaces", Housing and Home Finance Agency, Div. of House Research, Paper 32, April 1954.

DISCUSSION

JOHN STALEY, Louisville, Ky. (Written): Essentially all of the statements in evaluations are based on the ratio U_{1}/U. Accordingly, many of the statements, should they be taken out of context and without this understanding, can be misinterpreted. For example, "There is no apparent difference between the effect of Kraft or foil surfaces". This, of course, refers to the effect on the above ratio and not the absolute values of heat transmission. More correctly, the paper should be titled, "A Comparison of the Heat Transfer Characteristics of Mineral Wool Insulation with Reflective Surfaces as Given in the 1960 ASHRAE GUIDE to Experimentally Determined Values".

The experimental conditions chosen tend to minimize the effect of heat transmission by radiation compared to most actual construction conditions. Accordingly, the effect of reflective surfaces on the heat transfer has been reduced.

With regard to the effect of exposure to

attic dust on the emissivity of aluminum foil and the effect of such changes on heat transfer, it should be remembered that normally only one of two (in the case of mineral wool insulation combined with reflective surfaces) or one of four such surfaces are thus affected. The effectiveness of the remaining surfaces is unchanged.

The statement is made, "The increase in emissivity reduces foil effectiveness to a greater degree as the insulation thickness is increased." By similar analogy, the figures in Table XIII show that the second two inches of mineral wool insulation is only 18.1% as effective as the first two inches.

AUTHOR LUND: The paper covers the overall effectiveness of reflective surfaces in combination with mineral wool insulation which may be expected under field application. The absolute values must be considered in combination with other factors which are present as discussed in the paper. Mr. Staley does not mention the configuration of reflective insulations as shown in Fig. 8 which introduces a considerable error in the calculated transmittance when the absolute values are used. With regard to surface contamination of reflective surfaces upon exposure to atmospheric conditions, the results as shown are confined to mineral wool insulation with reflective surfaces. References to four surfaces apply to reflective insulations only where the combination of one surface contaminated and the configuration factor will further increase the error in the heat transmittance coefficient that may be expected. With reference to the last paragraph, it has long been recognized that the law of diminishing returns exists as the insulation thickness is increased.

F. A. JOY, University Park, Pa. (Written): The authors have investigated a wide range of construction in which air spaces have insulating value. One of these constructions—a 156 sq ft ceiling and attic combination—was studied at Pennsylvania State University in 1957 and the results were presented to the Society at the January 1958 meeting.

Comparing Professor Lund's Panel 45 with our Test No. 17 (each with a Kraft breather under a gable roof), he has ¼ in. less gypsum and 2 in. more joist depth, small differences that are partly offsetting. His tests, as shown in Table VIII, were made with the same 75 F room temperature that we used. He has two roof temperatures, 130 and 160 F. Interpolating to obtain a result for 150 F roof temperature, his heat flow rate is 22% higher than ours. With foil breather on Panel 48, his result is 23% more than ours, a notable consistency of disagreement.

At the same time, it was found by interpolation that his temperature difference across Panel 45 (Kraft) is only 2% higher than we observed, and across Panel 48 (foil) it is 5% higher. These small differences appear to be inconsistent with the higher heat flow reported.

Has Professor Lund made these comparisons and is there an explanation for this 22% difference?

On the same construction, tested with heat flow upon the guarded hotbox a comparison of Panel 40 (having a foil breather) can be compared with our Test No. 34. In this case, comparison must be made of "conductance" as used in this report. When corrected to 50 F mean temperature in both cases, Professor Lund's result is 10% higher than ours. This difference is not surprising for upward heat flow in consideration of the lower density insulation was used. It is unfortunate that the heat flow meter result is not also reported in this test.

Our experience with heat flow meters strongly indicates that they should be calibrated in contact with the same materials they will touch in service and that materials other than air are necessary.

Professor Lund has shown that a calculation of heat transfer made in accordance with GUIDE data is sometimes 25% lower than tests on well-built panels containing air spaces. He has also shown in an earlier work that joist effects are not properly determined in these cases by Fig. 7 of Chapter 9. Rather than applying a correction factor to the overall U value, it would appear preferable to use a true value for the joist effect; is this not feasible?

AUTHOR LUND: With regard on Professor Joy's comments on similar tests which he conducted on heat flow down with the results lower than we obtained, there are a number of factors which may contribute to this difference.

1. Professor Joy used 2 x 6 in. joists as compared with 2 x 8 in. joists in our tests. With 2 in. of insulation, 3⅝ in. of joist face exposure exists for the 6-in. joists whereas 5⅝ in. joist face exposure exists for 8-in. joists or increase of 155% in face area.

2. Our test area is 44 sq ft whereas his test area is 156 sq ft.

3. The slope of the gable on Professor Joy's tests is 5½ in. rise per foot whereas ours is 8¾ in. per foot. In addition Professor Joy has a 2 ft. sidewall in the attic between the ceiling and rafters of the roof.

4. Roof surface temperatures, radiation factors. influenced by the angle of the roof slope are additional variables to be considered.

5. The heat flow meters as stated in the paper were calibrated in contact with the insulation materials.

6. It should be noted that for heat flow up and heat flow horizontal our results agree very closely with the calculated values recommended by the GUIDE within the limits which may be expected.

7. With reference to Professor Joy's last paragraph, the use of a "true value" as he has suggested would be of considerable assistance in arriving at the overall transmittance for a particular construction. Possibly a "performance conductance value" which will include the studs or joists and insulation only for different directions of heat flow and

different mean temperatures would be a means of attaining this purpose.

C. F. PRIMMORE, New Kensington, Pa. (Written): The authors appear to have confirmed a number of factors that enter into the insulating value of air spaces and surfaces with emphasis on ceiling assemblies during summer conditions.

The conclusions reached in this paper are similar to those reached in an earlier paper by Chung and Lund published in December, 1959, i.e., that conventional methods of calculating values for heat flow down involving air spaces and surfaces may not be adequate. While this premise has merit under the conditions of test in this paper, it is questionable whether the margin of error is as significant as reported; even then it is only applicable to the specific method of installation considered in the tests.

This question arises because of certain inconsistencies in the data, plus reporting a margin of error that is assumed to be generally applicable where actually a simple change in the method of installation would provide quite different values.

The authors apparently neglected the consideration of evaluating alternate means of application to overcome the convective effects created by the joists.

The conclusions reached in this paper in regard to reflective surfaces with summer conditions are based on this surface being on the cold side where it has the opportunity to reflect to the joists. There are many methods of application where the reflective surface is on the warm side of an air space. An example would be an application where a reflective vapor barrier on mineral wool is inset stapled to provide an air space adjacent to the gypsum ceiling. Depending on the product involved, the reflective flanges could cover the joists. Certainly, in this instance the same margin of error reported in the paper would not have been shown. The advisability of the reflective surface being on the warm side of an air space for heat flow down was amply demonstrated in BMS 151, test 96 (referenced in paper).

The trend of merchandising extremely low density mineral wools has been evidenced for some time and this availability suggests the possibility of installing mineral wool flush with the top of ceiling joists. Actually, some contractors follow this as good practice today. This method of application should also reduce the joist effect on a reflective breather surface facing the attic. It is practical even to consider the manufacture of a reversed flange product for ceilings having a gabled roof. The flanges being on the breather side would cover the top of the joists. It is conceivable that a reflective breather material could be installed from a roll over top of the joists and mineral wool. None of the data in the paper is applicable to any of these improved modes of application.

This paper could lead to a misinterpretation so that the real value of reflective spaces and surfaces under all conditions of application might be overlooked. There is nothing wrong with the value of reflective spaces; the real cause and contributor to the margin of error reported in the paper is the ceiling joist. Instead of retarding interest in reflective surfaces, it is hoped that this paper can prove to be a valuable effort to stimulate a new study of application techniques.

In regard to the specific data reported, the authors have emphasized that their tests on air spaces for heat flow down do not agree with the values in the ASHRAE GUIDE. It is well known that the air space values in the GUIDE are based on guarded hotbox tests conducted at the Bureau of Standards. What this really means is that the guarded box utilized by the authors does not agree with the one at the Bureau of Standards. The authors were probably aware of this difference quite early in their evaluation. Did they consider running an exact duplicate of any of the Bureau tests to aid in finding an explanation?

Some of the differences found could undoubtedly be attributed to the change in the thermal conductivity of the mineral wool due to the mean temperature used in the tests. It is a well-known fact that heat flow through mineral wool increases as the mean temperature increases. Professor Lund stated in his presentation that the ASHRAE GUIDE calculated values assumed a mean temperature of 75 F for the wool because this was the only value available in the 1960 volume. This fact was not made adequately clear in the paper and thus the impression is created that any difference expressed was due to a significant margin of error in calculating reflective space values. The paper reports mineral wool thermal conductivity values of 0.27 at 75 F mean and 0.31 or more at 125 F mean. This results in a 0.48 change in thermal resistance per in. of mineral wool or a total change in resistance of 1.44 for a 3-in. thickness.

As mentioned earlier, the data appear to be inconsistent under some conditions. The authors stress the fact that the mineral wool used possessed a consistent thermal conductivity for material received over a two-year period. Yet when the conductance of Panel 13 is closely considered, it is equivalent to 6 in. of mineral wool with a mean temperature of 75 F as given in the latest volume of the ASHRAE GUIDE; or if the thermal conductivity is increased to 0.29 so as to correspond to the mean temperature in Panel 13, the conductance reported is equivalent to 6 in. of mineral wool plus the plywood and gypsum skins. This means, in effect, that the joist correction is equivalent to the value of the reflective space or more. This becomes hard to reconcile when one realizes that the air space conductance without joist effect in the ASHRAE GUIDE is 0.20—equivalent to a resistance of 5.00.

In regard to that portion of the paper dealing with the effect of attic dust on emissivity of aluminum foil, was any attempt made to have the reported values in the paper

represent actual integrated values for total attic areas? Attic locations near the louvers will be the most contaminated while more remote areas may be relatively clean. Locations in the air stream between louvers should also be more affected. What difference did the type and number of louvers have on the values? What difference did orientation have? Do the values reported represent integrated values for all these conditions?

AUTHOR LUND: Mr. Pridmore, in his discussion, includes many variables which are not within the scope of the study contained in our paper. The methods of installing the insulation and the reflective materials used in our studies are those which are predominately used in building practices. The authors acknowledge that there are many variations of this type of general application and agree that certain methods are superior to those which are generally followed, however, the question of economics is a factor. For example, inset stapled insulation provides an air space above the ceiling which is preferred as an additional air space is obtained. However, insofar as a vapor barrier is concerned, precautions must be taken that such inset stapling should include an overlap on the joist edges to insure sealing against vapor movement. It is also recognized that the advisability of reflective surfaces on the warm side of an air space for heat flow down as demonstrated in BMS 151 is good practice, but is very seldom followed in actual construction practices. Likewise, installing mineral wool flush with the top of ceiling joist is also good practice in reducing the joist effect on reflective breather surface facing the attic but again this is more of the exception that the rule in the building trades.

With regard to the misinterpretation of the real value of reflective spaces, this paper is confined to the results of the studies made within a limited scope without considering the commercial interests. With regard to the guarded hotbox tests conducted at the Bureau of Standards covering air space evaluation, the depth of air space was limited and less than those in some of our studies. In order to evaluate the air space coefficients from B.S. studies it is necessary to extrapolate the values to the thickness desired. Further review of these curves indicates that an average curve has been constructed through the test points. We are aware that further studies should be made of air space coefficients of greater depth or thickness.

Regarding the tests on the effect of attic dust on emissivity of aluminum foil, all samples were located in the central part of the attic remote from any louvres or openings to the outside. The authors have stated in the paper that additional studies should be conducted on this subject. It is also acknowledged in the paper that these studies were limited and that obviously further work should be done.

Greater emphasis covering the effect of joists on the overall coefficient of heat transmittance would have assisted in clarifying the results of these studies. It should be noted that the results obtained for heat flow horizontal and heat flow up are very close to those obtained by calculation from the GUIDE values. The principal disagreement occurs for heat flow down under summer conditions where higher temperature conditions exist across the ceiling and the effect of heat flow through the joists is greater.

1741

No. 1741

Thermal Effects of Floor Construction

RICHARD D. CRAMER LOREN W. NEUBAUER

During recent years, the California Experiment Station has been using experimental structures to investigate the impact of variations among micro climates on thermal conditions inside buildings. The initial structure used was a stock wood-frame house trailer which approximated the thermal characteristics of conventional residential construction. This work has been reported in the Journal of Home Economics.[1] Subsequently two additional structures were designed and built to reflect more nearly the characteristics of contemportary houses. These are called cubicles because they are in fact cubes 8 ft on a side. One wall is glass; the roof and solid walls are made of plywood skins over a light wood frame and are insulated with two in. of mineral wool. The floor is plywood, uninsulated. A concrete slab floor can be substituted for the wood floor, and this paper is concerned with a comparison of the influences of floor construction on the thermal conditions in otherwise identically detailed and oriented structures.

The cubicles have been used to study the effects of orientation on unprotected glass structures and of shading and screening devices, in the attempt to reduce radiant transmission through glass walls. This work was reported upon before this Society in 1959.[2]

The project is a part of a cooperative regional project sponsored by the Department of Agriculture and at Davis has the active participation of an architect, an engineer, and a horticulturist, representing coordinated efforts of the Departments of Home Economics, Agricultural Engineering, and Landscape Horticulture in the College of Agriculture. Initially the objective of the project was to evaluate methods for alleviation of

1. Journal of Home Economics, Vol. 50, No. 3, March 1958, pp. 181-184.
2. ASHRAE Transactions, Vol. 65, 1959, page 499.

Richard D. Cramer is Associate Professor of Housing, Dept. of Home Economics and Loren W. Neubauer is Associate Professor, Dept. of Agricultural Engineering, University of California, Davis. This paper was prepared for presentation at the ASHRAE Semiannual Meeting, Chicago, Ill., February 13-16, 1961.

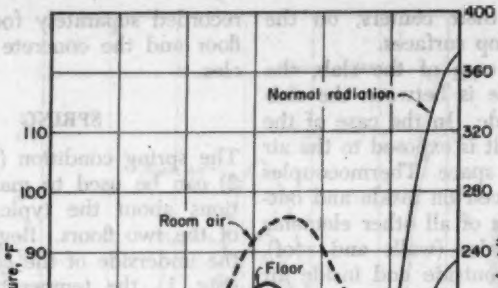


Fig. 1 Spring equinox — concrete floor cubicle temperature

temperature extremes in hot, dry summers which are typical of the California central valleys and of much of the southwest. In this kind of climate, summer cooling is a more difficult problem to solve than winter heating. Midafternoon temperatures of 110 F are not unknown, 105 F not uncommon. At the same time night-time temperatures below 70 F are normal, making a 40 F diurnal spread typical. Hot days are cloudless and relatively still. Humidity at mid-afternoon is usually lower than 20%. With the rapid fall of temperature in the early evening come breezes from the south which make the evenings comfortable.

During portions of 1957 and 1958, two cubicles were operated simultaneously, with the glass wall facing south in an exposed field without shade except for a bit on the east and west horizons, and with the glass wall otherwise unprotected, in an attempt to assess the contributions of floor construction to interior thermal conditions. Specifically, the floors compared were the wood floor previously described which has a six-in. air space beneath, open to the atmosphere at its periphery, and a concrete slab poured on grade. Neither floor is insulated. Both have thermo-

couples in their centers, on the under and top surfaces.

In the case of the slab, the thermocouple is between the slab and the grade. In the case of the wood floor, it is exposed to the air of the crawl space. Thermocouples are also placed on inside and outside surfaces of all other elements of the cubicles (walls and roof), and in the outside and inside air shielded from radiation.

Measuring and recording is done with an instrument which takes a reading from each thermocouple every six min and records on a running chart. Thus, 24-hr records are available for each thermocouple, and each of the results used herein represents the average of at least three typically hot, dry, cloudless days. These tests have been performed at the solstices and at the equinoxes. Since clear weather was necessary, the clear days which occurred nearest the solstice or equinox were those used. Radiation normal to the sun, derived from pyrheliometer measurements, has been included in order to indicate that the four periods were comparable. Radiation on a south-facing vertical surface is shown as an indication of the energy to which the glass wall was exposed.

Soil temperatures four in. below the surface in an unsheltered location were taken from Davis weather station records and are included for comparison with temperatures under four in. of concrete – the cubicle floor. The results, then, can be termed spring, summer, fall, and winter conditions recorded separately for the wood floor and the concrete floor cubicles.

SPRING

The spring condition (Figs. 1 and 2) can be used to make observations about the typical behavior of the two floors. Beginning with the underside of the concrete slab (Fig. 1), the temperature is relatively constant throughout the 24-hr period, varying 5 F, with the low point at 10:00 in the morning and a high point in the early evening. The high point is the result of exposure of sizable proportions of the slab to radiation which penetrated the glass wall. The equinox sun has a vertical angle of about 51 deg at noon in this latitude. Resulting heat stored in the slab to some extent penetrated through the depth of the slab and warmed its under surface. A comparison of the temperature of the under surface and the temperature of the air in the room, indicates that the room air is cooler from 10:00 p.m. until 8:00 a.m. Therefore, the room air gains heat from the slab, or the slab loses heat to the room at night. However, from 8:00 a.m. until 10:00 p.m., the slab gains heat from the room air. That portion, which is not exposed to the sun, cools the room air by receiving heat from it. The portion in the sun gives off less heat to the air than does the wood floor, because heat flows from the concrete surface into the slab at a faster rate than it flows through the plywood floor. If the glass were shaded or if it faced North, the slab would not receive direct

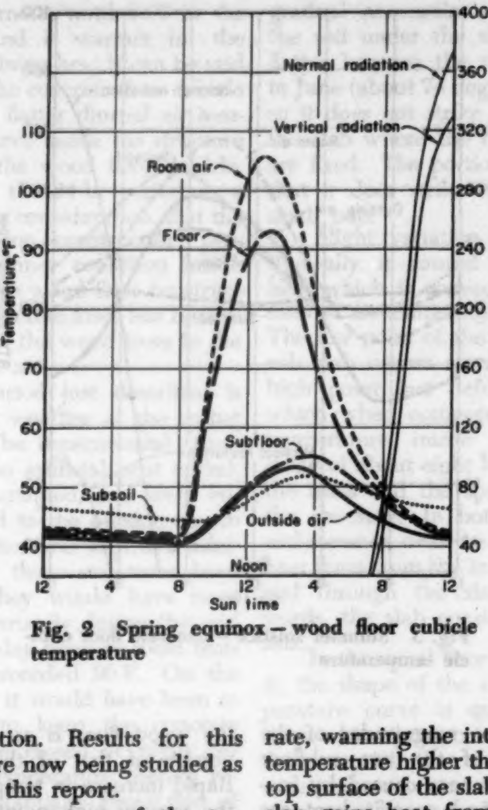


Fig. 2 Spring equinox – wood floor cubicle temperature

solar radiation. Results for this condition are now being studied as a sequel to this report.

The temperature of the top surface of the slab describes a curve which shows the radiant gain resulting from solar impingement on a portion of its surface including the thermocouple location. During the daytime hrs, the room temperature rises above that of the slab by virtue of radiation through the glass impinging on interior wall surfaces. These are plywood, rather dark in color. Because they are backed with insulation, they give off heat at a comparatively rapid rate, warming the interior air to a temperature higher than that of the top surface of the slab. And so the slab gains heat from the interior air by convection over its entire surface, and from the sun by radiation over a portion of its surface.

For the wood floor cubicle (Fig. 2), the temperature curve for the underside of the floor is variable during the 24-hr period, as compared with the corresponding concrete slab curve. The low point of its curve occurs at 6:00 a.m. and the high point at 3:00 p.m., and the spread is somewhat greater than 20 F. Comparing both the

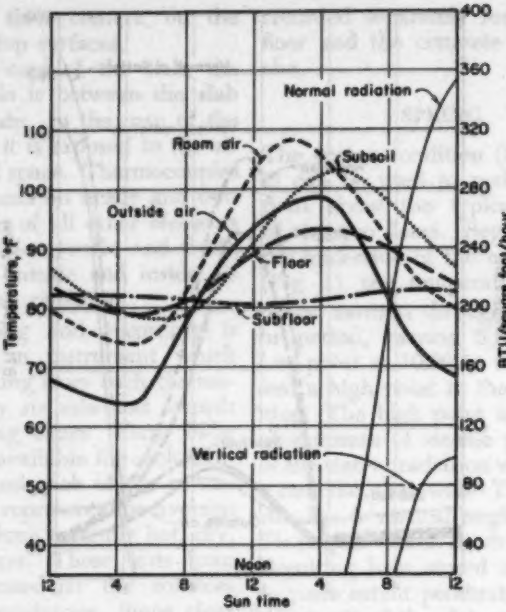


Fig. 3 Summer solstice — concrete floor cubicle temperature

times and the magnitudes of the high points of the two subfloor curves, differences caused by material mass and specific heat are apparent, a delayed response in the case of concrete, a rather quick response in the case of wood. Average value of the wood curve is lower than that of the concrete curve. The area between the curves at night is substantially larger than the area between the curves during the day, making the wood floor cubicle on the whole cooler, and conversely, the concrete floor cubicle warmer.

Behavior of the top surface of the wood floor is an exaggerated version of that of the underside. Rapid increase in temperature and the narrow high point are caused by direct solar radiation through the glass onto the floor surface. In comparison with concrete, the value is higher during mid-day, and lower at night. The top surface of the wood floor is cooler than the top surface of the concrete floor for substantially more than 50% of the time.

Room temperature in the wood floor cubicle is cooler than the room temperature in the cubicle with the concrete floor from 4:00

in the afternoon until 9:00 in the morning, and is warmer for the seven remaining hrs. It can be said then, that the concrete floor cubicle produces a flatter diurnal air temperature curve inside the structure than does the wood floor cubicle, and taking the 24-hr period as a whole under consideration, that the concrete floor construction produces a warmer condition inside than does the wood floor construction. The concrete loses less heat to the soil than the wood loses to the crawl space air.

The period just described is warm clear weather at the spring equinox. The experimental structures had no artificial heat added, were not occupied, and were entirely closed to the outside air, so that infiltration was kept to a minimum. Had these structures been inhabited they would have been opened generously during the mid part of the day because room temperatures exceeded 90 F. On the other hand, it would have been to advantage to keep the concrete floor cubicle closed at night because the temperature therein remained substantially above that of the outside air. In the case of the wood floor cubicle, night-time temperatures inside and out were almost identical, and so it would make little difference whether it were open or closed.

SUMMER

On the summer solstice (Figs. 3 and 4), the temperature curve for the underside of the slab (Fig. 3) is both higher and flatter than the corresponding curve in spring. It is higher because there has been a gradual accumulation of heat in the soil under the slab, and it is flatter because the sun is steeper in June (about 74 deg at noon), and so it does not strike the center of the slab where the thermocouples are fixed. The portion of the slab that it does strike is a relatively small one.

Slight variation in the curve diurnally is caused primarily by heat which is convected from the interior air temperature to the slab. The low point of the curve for the sub slab occurs at noon, and the high point just before midnight, which when compared to the air temperature inside illustrates a delay of about eight hrs, caused by the mass and the specific heat of the concrete. In both the spring and summer, over the 24-hr period, heat flows from the interior air into and through the slab. In other words, the slab constitutes a heat sink.

In the wood floor cubicle (Fig. 4), the shape of the sub-floor temperature curve is quite like the shape of its spring counterpart with the exception that, in comparison with the concrete sub-floor temperature, the wood sub-floor is proportionately (over the 24-hr period) warmer in June than in March, or conversely, concrete is proportionately cooler. The relationships between outside air temperature and wood sub-floor temperature are similar March and June.

However, mid-day floor and room temperatures have reversed positions in June as compared with March, again because of steeper sun angles. For the same reason there is a smaller difference in June

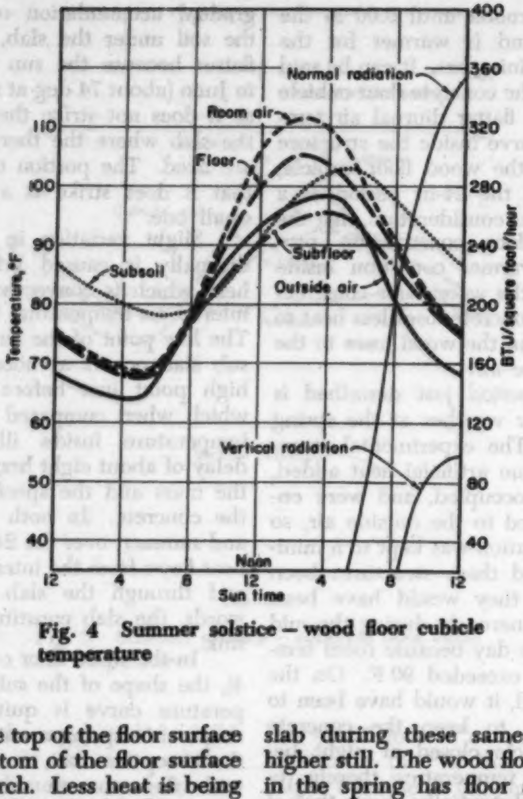


Fig. 4 Summer solstice – wood floor cubicle temperature

between the top of the floor surface and the bottom of the floor surface than in March. Less heat is being transferred through the floor construction to the crawl space underneath.

In the spring, during the night, wood floor cubicle temperatures – interior air, and the floor surfaces – are similar to the temperature of the outside air. In the concrete floor cubicle the top side of the slab and the interior air temperature are consistently warmer than the outside air through the middle of the night and early morning hrs, and the underside of the concrete slab during these same hours is higher still. The wood floor cubicle in the spring has floor and room temperatures coinciding with outside air at night, but in the summer they remain above the outside air temperature. Perhaps the reason for the spread in June as compared with March can be found in the greater diurnal spread of the outside air temperature curve itself, and in the consequent slope of the outside air temperature change which produces a lag in inside temperature changes. Looking ahead for a moment to the fall, there is exaggerated example of

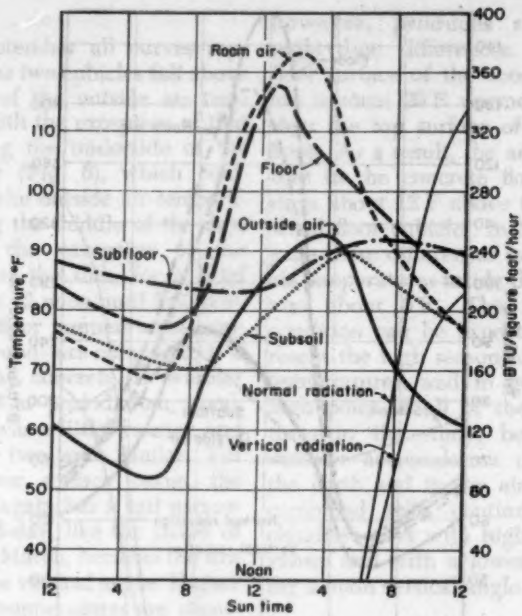


Fig. 5 Fall equinox — concrete floor cubicle temperature

the summer condition, probably for the same reason.

FALL

Turning to September (Figs. 5 and 6), there are patterns in some ways comparable to March and in some ways to June. They are comparable to March in that again substantial amounts of direct radiation penetrate the cubicle and impinge on floor and wall surfaces, but at the same time the outside air temperatures are higher, a condition more like June. As a result, higher inside air and floor surface temperatures are experienced during the fall.

For the underside of the concrete slab (Fig. 5), the fall curve which again is higher than its predecessor, has returned because of radiation to a more markedly curved shape as opposed to the flatter shape observed in June. It is a sharper curve than it was in March, perhaps reflecting a steeper outside air temperature. It is known that concrete slabs lose heat laterally to the periphery in contact with the outside air.

It may also be true that, because temperature differentials be-

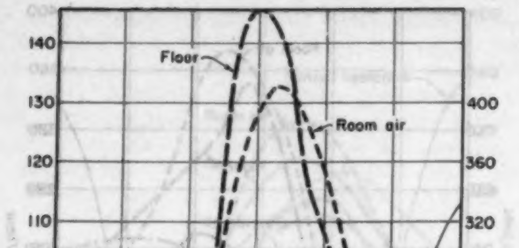


Fig. 6 Fall equinox – wood floor cubicle temperature

tween slab and soil are greater, there is a faster rate of heat transfer from the underside of the slab into the soil than was the case in March. At any rate, the low point again falls at 11:00 a.m., near the spring low point, and the high point at about 7:00 p.m., again approximating the March condition. Viewing the sub-slab temperature in relation to the outside air temperature, however, in comparison with June, the average temperature of the underside of the slab has become progressively warmer than the average outside air temperature, whereas in June they were nearly the same.

In June the sub-slab temperature is below that of the outside air during most of the daylight hours and above at night, the two averaging about the same diurnal value. In September, on the other hand, the average temperature of the sub-slab is well above that of the outside air. As the summer passes the slab warms progres-

sively.

In September all curves representing the two cubicles fall above the curve of the outside air temperature with the exception of that representing the underside of the wood floor (Fig. 6), which coincides with the outside air temperature during the middle of the day, and with the exception of the underside of the slab, for a brief period from 12 noon until 5:00 p.m. When sub-floor temperatures—concrete and wood—are compared, it is evident that concrete is warmer over the 24-hr period than wood. In April it was a little warmer, and in June the two were similar. For the top floor surface curve, the wood floor again has a tall narrow peak at mid-day, like the shape of the peak in March, because the sun is at the same vertical angle. Higher interior air temperatures are stimulated by higher outside air temperatures, whereby the rate of cooling by conduction through the glass wall and the other surfaces is slower.

The differences between the concrete slab top surface and the wood top surface temperatures are much the same as they were in March. Both curves have been raised correspondingly. Interior air temperature curves are similar in shape and in proportion to those in March, but they reveal a slightly greater difference between the wood floor cubicle air temperature and the concrete floor cubicle air temperature during the middle of the day.

The 40 F diurnal spread in the outside air temperature curve, however, produces substantial night-time differences. The top floor surface of the concrete cubicle is some 20 F warmer at night than the top surface of the wood floor. As a result, the air temperature in the concrete floor cubicle stays about 12 F above that of the wood floor cubicle. In March and June these differences in nocturnal air temperatures inside the cubicles was about 6 F. The September condition can be expected to represent the high season for sub-slab temperatures, and in general the high point of all of the temperatures in the study, because the summer accumulation of heat in the earth and in the air mass are combined with continuing clear cloudless days with high radiation values and with a lower sun having a noon vertical angle of 51 deg.

WINTER

Toward the end of December (Figs. 7 and 8), the sub-slab curve (Fig. 7) is shaped quite as is the corresponding curve in September. The entire slab is exposed to radiation at noon; however, the center of the slab, where temperatures were measured, had full sunshine in September, and again in December, and so the shapes are similar. In general, there has been a rapid decline in sub-slab temperature from an average of about 88 F in September to an average of about 59 F in December. This is the period of the year when the most rapid change in sub-slab temperature takes place, because from December through March and into June and then September, there is

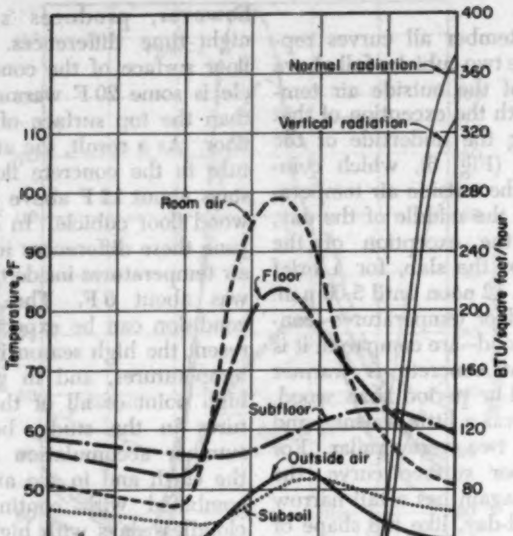


Fig. 7 Winter solstice—concrete floor cubicle temperature

a gradual rise representing a period of nine months. The corresponding decline requires three months. However, December may not be the low point. It probably occurs in January or February more nearly evening out the gain and loss periods.

In December for the first time, the sub-slab temperature is substantially higher than the outdoor air temperature during the entire 24-hr period. Heat flows upward from the ground through the slab into the air of the cubicle from 6:30 p.m. until 9:00 a.m. As in September sub-floor temperatures of the wood floor (Fig. 8) are about the same as the outside air temperature during the early part of the day and higher during the latter part of the day.

The curves representing the top surfaces of concrete and wood floors, respectively, bear some resemblance to their counterparts in September, except that the wood floor top surface in December is only slightly warmer than the concrete floor top surface during the early afternoon hours, whereas the concrete floor surface is a good deal warmer than the wood floor surface during the rest of the day.

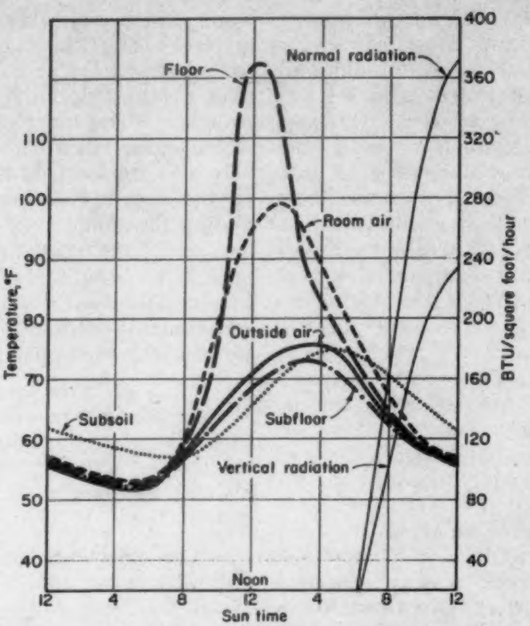


Fig. 8 Winter solstice — wood floor cubicle temperature

This change presumably has been brought about by the relatively warmer concrete sub-floor which initiates heat transfer upward through the slab.

The relationship of interior air temperature curves, concrete to wood, is much like that noted in March and something like that in September but with a smaller difference. The mid-day concrete floor cubicle is cooler by a smaller margin than the wood floor cubicle than is the case in September.

At night in December and in March the outside air clusters with the wood curves (sub-floor, floor, and room air). During June and September the three wood curves remain together, but the temperature of the outside air which is in contact with the crawl space air, is a little lower than the cluster of wood floor cubicle curves, indicating a more rapid transfer from the floor through the crawl space to the outside air than in December and March.

In summary, because the only difference between the two cubicles lies in the floor construction, the annual behavior of the subfloor temperatures stimulates the differences in interior air tempera-

ture patterns. There are smaller differences and flatter curves for the concrete floor cubicle and greater differences and steeper curves for the wood floor cubicle. The concrete sub-floor varies from a minimum of about 55 F at 10:30 a.m. in December to a maximum of about 92 F at about 7:00 p.m. in September. The wood sub-floor varies from a minimum of 40 F at 8:00 a.m. in December (and is undoubtedly lower on colder days) to a maximum of 91 F at 3:30 p.m. in September. The wood floor has a six-month cycle of warming and cooling. The concrete floor has probably an eight-month cycle of warming and a four-month cycle of cooling.

As a result, in March and again in June interior air temperatures in the concrete floor cubicle are 5 or 6 F warmer than those in the wood floor cubicle during the early hrs of the morning. This difference is increased to about 12 F in September and then decreased to about 8 F in December. During the middle of the day, March air temperatures inside the wood cubicle are about 5 F warmer than those in the concrete cubicle, in June 3 F warmer, in September about 10 F, and in December about 5 F.

For purposes of evaluation, then, it can be said that concrete slabs have the advantage of being cooler during the hot part of the day. On the other hand, the concrete slab is warmer at night. A warm floor is clearly an advantage in December with its cold nighttime temperatures, colder than those comfortable inside inhabited structures. The same can be said of the spring when nocturnal temperatures are too cold for comfort in the wood floor construction. During the summer, in both June and in September, the wood floor cubicle has an advantage at night, but its advantage during this season could be virtually eliminated by generous nocturnal ventilation.

All in all, it can be said that in the climate in which these experiments are being conducted there are thermal advantages to concrete slab construction over wood floor construction.

The objection may be raised that the crawl space under the wood floor was lower than that normally encountered; that it was not enclosed around its periphery; that ventilation exceeded that which is encountered in a normally ventilated crawl space; and that it is therefore probable that the differences encountered between slab and wood floor construction would be somewhat reduced, if standard house construction had been used instead of the experimental construction of the crawl space involved. However, the height of the air space and the degree of ventilation tend to counteract each other, so probably the difference would not be significant.

DISCUSSION

C. W. POLLOCK, Johnstown, Pa.: Was the general purpose of this investigation to determine the type of construction that is most desirable?

AUTHOR CRAMER: Yes, concrete floors are being used quite commonly in house construction in California. Comparable repetition with glass facing north to eliminate the effect of direct radiation on the slab has also been done. Results in this paper are strongly influenced by the impingement of the sun on the slab.

M. W. KEYES, Pittsburgh, Pa.: Would a carpet covering substantially all of the concrete give results similar to those for wood?

AUTHOR CRAMER: It would give results closer to those for wood than the results indicated here and, of course, it would depend on the insulating effect of the specific carpet involved.

No. 1742

Evaluation of Wet-Bulb Data for Cooling Equipment Design

LOREN W. CROW
Member ASHRAE

The capacity and efficiency of any cooling tower which uses the unmodified atmosphere as an agent for carrying off heat must depend on the characteristics of the atmosphere at the particular point on the globe where the cooling tower is located.

It will be assumed here that the reader is familiar with how wet-bulb temperatures are measured. It will also be assumed that the reader understands that any scale of wet-bulb values is approximately the same as the enthalpy scale on a psychrometric diagram.

Design engineers who make decisions related to buying, building, or selling evaporative cooling towers that operationally must deliver cooled fluid of not more than some specified maximum temperature must concern themselves with the variable nature of the atmosphere. A decision in economics is required for each particular installation. If the cooling tower is larger than required, its cost will be unnecessarily high. If the cooling tower is too small, it will not accomplish its assigned task and heavy monetary losses may result.

Cooling towers are built to last for several years, and their ultimate capacity is tested during the summer months when wet-bulb temperatures reach their highest readings at any particular locality. However, the temperature sequence is not the same in any two summers. Fig. 1 shows in a general way the relationship between fluctuating wet-bulb temperatures and design specifications.

In the historical development of cooling tower installation it is unfortunately true that personal points of view rather than basic detailed wet-bulb frequency data have often influenced the selection

Loren W. Crow is a Consulting Meterologist. This paper was prepared for presentation at the ASHRAE Semiannual Meeting, Chicago, Ill., February 13-16, 1961.

of design criteria. With a client who is willing to pay the bill, it is only natural that a conservative consulting engineer will tend to over-design.

If there are instances of radical over-design, it is only natural that a careful engineer employed by a cooling tower manufacturer will find ways to save his employer money and still produce and install a completely adequate tower.

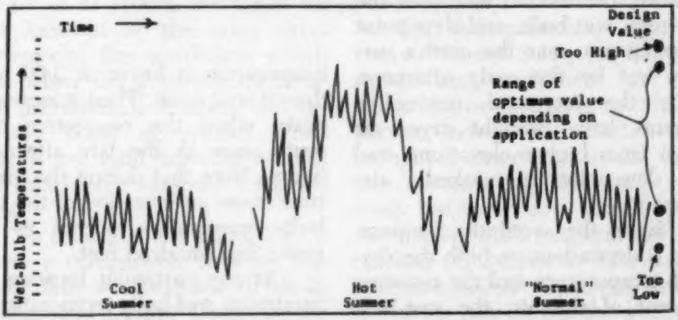
Should the manufacturer adjust downward by a similar factor, the specifications of another more careful and precise consulting engineer, there is great danger of producing a tower which will fail. However, any such failure may not become apparent for 2 or 3 years after the tower is placed in service. The first summer, or even two summers following installation, may be abnormally cool. (See Fig. 1.)

In order to furnish all engineers with adequate basic data for selecting proper design wet-bulb values one manufacturer of cooling towers has supported an extensive tabulation* of detailed wet-bulb values. Over ten million hourly weather observations have been counted by electronic tabulation methods to develop the actual frequency, by degrees, of wet-bulb temperatures at 396 locations within the United States and 36 foreign locations. The time period of observations was during the ten-year span, 1948 through 1957. The material in the published book comprises the most comprehensive treatment of summer wet-bulb temperature frequencies available, and it is based on actual observations made by trained observers.

It is hoped that the expanding use of the published material will lead to more careful selection of

* A summarization of the tabulations can be found in the book, *Evaluated Weather Data for Cooling Equipment Design.* Fluor Products Company.

Fig. 1 Illustration of variable wet-bulb temperature sequences as related to selection of design criteria for cooling towers (summer time scale schematic only)



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Cooling towers are built to last for several years, and their ultimate capacity is tested during the summer months when wet-bulb temperatures reach their highest readings at any particular locality. However, the temperature sequence is not the same in any two summers. Fig. 1 shows in a general way the relationship between fluctuating wet-bulb temperatures and design specifications.

In the historical development of cooling tower installation it is unfortunately true that personal points of view rather than basic detailed wet-bulb frequency data have often influenced the selection

Loren W. Crow is a Consulting Meterologist. This paper was prepared for presentation at the ASHRAE Smiannual Meeting, Chicago, Ill., February 13-16, 1961.

of design criteria. With a client who is willing to pay the bill, it is only natural that a conservative consulting engineer will tend to over-design.

If there are instances of radical over-design, it is only natural that a careful engineer employed by a cooling tower manufacturer will find ways to save his employer money and still produce and install a completely adequate tower.

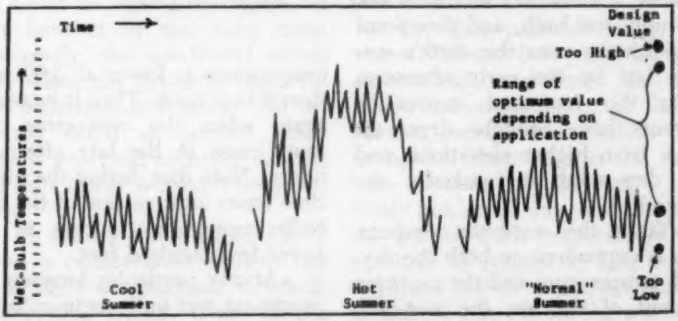
Should the manufacturer adjust downward by a similar factor, the specifications of another more careful and precise consulting engineer, there is great danger of producing a tower which will fail. However, any such failure may not become apparent for 2 or 3 years after the tower is placed in service. The first summer, or even two summers following installation, may be abnormally cool. (See Fig. 1.)

In order to furnish all engineers with adequate basic data for selecting proper design wet-bulb values one manufacturer of cooling towers has supported an extensive tabulation* of detailed wet-bulb values. Over ten million hourly weather observations have been counted by electronic tabulation methods to develop the actual frequency, by degrees, of wet-bulb temperatures at 396 locations within the United States and 36 foreign locations. The time period of observations was during the ten-year span, 1948 through 1957. The material in the published book comprises the most comprehensive treatment of summer wet-bulb temperature frequencies available, and it is based on actual observations made by trained observers.

It is hoped that the expanding use of the published material will lead to more careful selection of

* A summarization of the tabulations can be found in the book, *Evaluated Weather Data for Cooling Equipment Design.* Fluor Products Company.

Fig. 1 Illustration of variable wet-bulb temperature sequences as related to selection of design criteria for cooling towers (summer time scale schematic only)



design criteria and fewer and fewer mistakes which heretofore may have been due to lack of detailed weather data.

Those concerned with design problems may be interested in a brief review of some of the topical material covered in the book.

TEMPERATURE VARIATIONS

Weather sequences are such that any one element such as the wet-bulb temperature shows a broad range of variability with time. This variability is noticeable whether days, months, or seasons are being considered.

DAILY TEMPERATURE SEQUENCES

In Fig. 2 a series of graphs of dew-point, wet-bulb, and dry-bulb temperatures for the lower 1000 ft of the atmosphere are given to show a typical daily pattern of temperatures and lapse rates.

The small arrows at 9:00 a.m., noon, and 3:00 p.m. in Fig. 2 indicate the convection which is brought about by the heating of the air near the earth's surface. Initially, conduction increases the dry-bulb, wet-bulb, and dew-point temperatures near the earth's surface, but by the early afternoon hours the increased convective currents have brought dryer air down from higher elevations, and the dew-point is markedly decreased.

Since the wet-bulb temperature is dependent on both the dry-bulb temperature and the moisture content of the air, the wet-bulb temperature is lower at 3:00 p.m. than it is at noon. Then it increases again when the convective currents cease in the late afternoon hours. Note that during the nighttime hours an inversion of the dry-bulb temperature occurs in the lower few hundred feet.

At any particular location the maximum wet-bulb temperature is

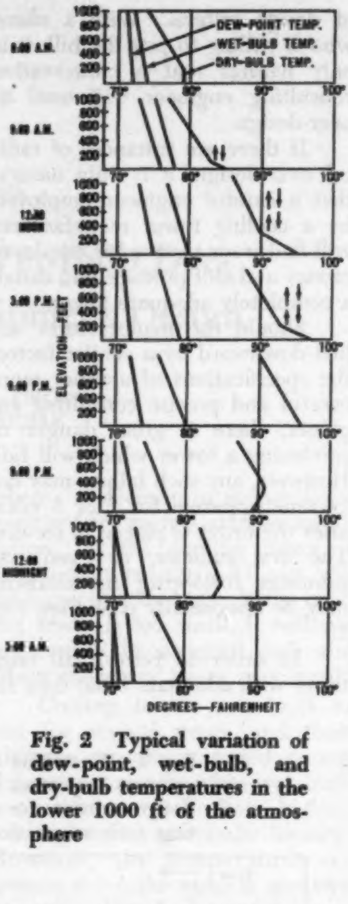


Fig. 2 Typical variation of dew-point, wet-bulb, and dry-bulb temperatures in the lower 1000 ft of the atmosphere

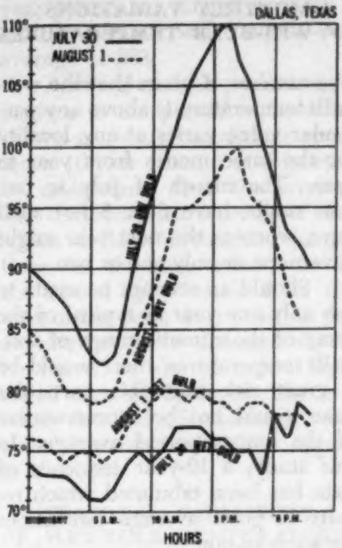


Fig. 3 Hourly sequence of dry-bulb and wet-bulb temperatures on hottest dry-bulb day and hottest wet-bulb day at Dallas, Texas, during 1957 summer season

reached at a time when there is high moisture content. The dry-bulb temperature generally would not be at its record maximum for that location at the same time. Conversely, the conditions which exist when the dry-bulb temperature maximum is reached will ordinarily occur at periods when there is less total moisture content and thus lower wet-bulb temperatures.

To illustrate this point, Fig. 3 shows a comparison between the actual daily wet-bulb and dry-bulb sequences at Dallas, Texas, on the peak wet-bulb and dry-bulb days in 1957. Somewhat by coincidence, these two days occurred only two days apart in the 1957 season.

The air mass which prevailed through the evening hours of July 30 was a dry, hot mass with air moving from the southwest or west over Dallas. At midnight the dry-bulb temperature was 90 F and the wet-bulb temperature was 74 F. The hourly sequence of wet-bulb and dry-bulb temperatures shows a decrease in both of these elements between midnight and 6:00 a.m. A rapid increase in both wet-bulb and dry-bulb temperatures took place from 6:00 a.m. until noon, except for a hesitation in the wet-bulb climb at 9:00 a.m., probably caused by convective currents carrying moisture aloft and bringing in dry air from above. By noon the maximum wet-bulb temperature was reached. This held steady for 2 hours as the dry-bulb temperature continued its movement upward. The peak dry-bulb temperature of 110 F was reached at 4:00 p.m., at which time the wet-bulb temperature had decreased by 1 F from its maximum, which prevailed from noon to 2:00 p.m.

On that particular day, between 7:00 and 8:00 p.m., the wind shifted and a different air mass which had originated in the Gulf of Mexico moved over Dallas. This brought a sharp rise in the moisture content. July 31 was a relatively hot, moist day, and the peak day for wet-bulb temperatures was reached on August 1. The hourly sequence shows a pattern similar to that for July 30, but the wet-bulb

temperatures are higher while the dry-bulbs are lower. The maximum wet-bulb temperature was reached on August 1 at 11:00 a.m. and continued high through 7:00 p.m.

The maximum dry-bulb temperature of 100 F occurred at 5:00 p.m., six hours after the maximum wet-bulb temperature had been recorded. The temperature fell rapidly from this peak, being only 95 F at 7:00 p.m. Both temperatures dropped from 7:00 p.m. to midnight. A comparison of the two extreme days shows that:

The maximum wet-bulb temperature at Dallas was reached when the dry-bulb temperature was between 90 and 95 F.

The actual maximum dry-bulb temperature on the hottest dry-bulb day was some 15 F higher than 95 F (110 F).

The peak dry-bulb temperature on the hottest dry-bulb day was reached when the wet-bulb temperature was 3 F less than it was on the peak wet-bulb day.

MONTHLY VARIATIONS OF WET-BULB TEMPERATURES

The number of hours that the wet-bulb temperature is above any particular value varies at any locality for the same month from year to year. The month of July in one year might have 6 or 8 hot spell days, whereas the next year might have none or only one or two.

Should an attempt be made to use only one year as typical of the array of the climatic range of wet-bulb temperatures, there would be a great risk that this particular year would not be representative of the longer period average. In this study, a 10-year sequence of data has been tabulated which results in good average indications for each station.

In Table I is shown the 10-year July record of high- wet-bulb temperature frequencies at Omaha, Nebraska.

The number of hours when the wet-bulb temperature was 73 F or more at Omaha during the 10-

TABLE I
TEN SEASONS OF JULY HIGH WET-BULB TEMPERATURES AT OMAHA, NEBRASKA

Wet Bulb,	Hours									
F	1948	1949	1950	1951	1952	1953	1954	1955	1956	1957
82					3					1
81					2		1		1	7
80	1	6		11	5	12	3		2	9
79	5	9		8	7	16	9	1	3	12
78	15	39		18	29	8	10	6	9	21
77	14	34		17	34	19	20	27	14	45
76	18	53	2	26	25	31	28	55	6	68
75	32	60	4	42	34	45	34	80	13	75
74	53	46	7	33	28	34	53	105	17	81
73	38	27	15	36	65	37	43	101	97	67
Totals	176	274	28	191	229	202	201	375	162	386

year period has ranged from 28 in 1950 to 386 in 1957. The 10-year average is 233.

Although values presented for stations in the published book are shown as average conditions covering a 10-year span, it is important to understand that the average was derived from a series of data which reflect noticeable fluctuations for the same month or for the season as a whole from one year to the next. It is possible that on rare occasions a design should be based on the recorded highs rather than the averages. The necessary detailed information is available, although it does not appear in the book.

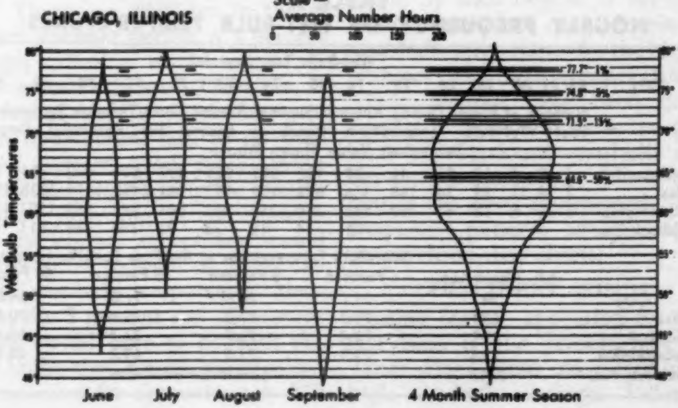
SEASONAL VARIATIONS OF WET-BULB TEMPERATURES

Although the summer period treated in the book covers the four months of June, July, August, and September, most of the notably high wet-bulb temperatures occur during the two months of July and August at most stations.

To illustrate the seasonal fluctuation in wet-bulb temperature frequency, the monthly pattern, which in turn produces the four-month seasonal total, is presented for five locations in the United States.

The arrays at Chicago, Ill., Fig. 4, illustrate the fact that nearly all the top wet-bulb temperatures in the four-month seasonal period actually were contributed during the months of July and August; virtually none of the hours in the upper 1 percent, and very few of those in the upper 5 per cent, were recorded in June and September.

Fig. 4 Average monthly and seasonal pattern of wet-bulb temperatures during the summer months of June through September at Chicago, Illinois



The lower end of the array for the four-month seasonal period is made up of hours contributed by the months of June and September. During these two months it is possible to have air masses over Chicago which are considerably cooler and dryer than those experienced in the peak warm period of July and August.

When low design values are so chosen that the entire population of such wet-bulb temperatures are not recorded in the four summer months, June through September, it would become necessary to estimate the numbers of such hours contributed by the months of May and October.

HOT SPELL WEATHER

If wet-bulb temperatures had the same large fluctuation in a 24-hr period as that which is common for dry-bulb temperatures, it would be possible in some types of installations for the cooling tower equipment to make up during the cooler night-time hours much of the loss occurring during the peak heat period of the day (a flywheel effect).

However, in the case of wet-bulb temperatures, it is somewhat typical that during hot spells the wet-bulb temperatures shows very little fluctuation during the 24-hr periods.

By examination of several hot spell sequences at various stations it was determined that if one moves down from the top to the 15th percentile level in the population of all the June through September hours, this will be a value which will be exceeded for one or more full 24-hr periods per summer season. Thus it was concluded that a low recommended design limit should be at the 15th percentile level of all hours, June through September.

TABLE II
HOURLY FREQUENCY OF WET-BULB TEMPERATURES

City	Wet-Bulb Temperatures 82	81	80	79	78	77	76	75	74	73	72	71	70	69	68
	Number of Hours During Average Summer Season (June Through September) That Wet-Bulb Temperatures Equal or Exceed the Particular Degree Fahrenheit Value Shown Above														
Chicago ...	1	2	4	10	21	46	79	131	207	286	387	504	632	751	889
Tulsa	2	4	11	29	74	165	306	532	753	993	1241	1452	1674	1835	1981
Philadelphia	1	2	6	15	40	87	148	240	351	483	641	787	950	1107	1256
Lubbock, Tex.							1	4	12	32	91	200	386	611	876

City	Alt. Above Sea, Ft	Wet-Bulb Temperatures at the Four Specific Frequency 1 Percent	5 Percent	15 Percent	50 Percent
		77.7	74.8	71.5	64.6
Tulsa........	674	79.0	77.2	75.4	70.9
Chicago.....	623	78.4	76.0	73.3	66.6
Philadelphia..	114	73.1	71.5	69.8	65.9
Lubbock, Tex.	3242	Levels Listed Below			

WET-BULB DESIGN VALUES

The primary purpose in the preparation of the book was to present the best available information in a manner that would be the most helpful to the engineer in choosing optimum design values.

As previously noted, the data upon which the reported information is based were obtained from the most reliable sources available. Hourly observations in the majority of cases were made continuously for a 10-year period. The portion representing the 4-month period, June, July, August, and September, has been analyzed and the results presented in such a way that the designer can choose the particular wet-bulb temperature for a given locality that will be above the wet-bulb temperatures of any given percentage of the total hours of the period. This wet-bulb temperature is more usefully designated as the value equalled or exceeded a given number of hours or a given percent of the total hours.

There are 2928 hrs in the 4-month summer period. A 1 percent design criterion would entail approximately 30 hrs. Therefore, a statement that a particular wet-bulb temperature had to do with the 1 percent level would mean that for any year during the months of June, July, August, and September the normal expectancy would be for 30 hrs to be at or above the noted wet-bulb temperature. Similarly, the 5 percent and 15 percent levels would have to do with the wet-bulbs that would be equalled or exceeded for approximately 150 hrs and 450 hrs, respectively.

The frequency array shown in Table II is a cumulative total of the number of hours at which the noted wet-bulb temperatures were equalled or exceeded, starting from the highest temperature and working downward through the entire sample of wet-bulb temperatures for each station.

Table II is an excerpt from a rather entensive table which is included in the book, showing the detailed tabulations at 432 locations.

In addition to the table, isoline maps of the three design levels, 1 percent, 5 percent, and 15 percent are presented.

COINCIDENT DRY-BULB VALUES

One of the completely new tabulations presented deals with the dry-bulb temperatures which occur coincident with high wet-bulb readings. At 49 locations scattered throughout the entire country, complete coincident arrays were made by each degree of dry-bulb temperature and each degree of wet-bulb temperature for the higher summer wet-bulb temperatures. These stations have been given special treatment to show the pattern of coincident wet-bulb and dry-bulb temperatures which prevail in the areas represented. The frequency patterns are portrayed on psychrometric charts.

COINCIDENT WINDS

Another feature appearing for the first time is the presentation of winds which occur coincident with high wet-bulb readings. Velocity

gradations and concentrations of direction are indicated for all stations throughout the United States.

INTERPOLATION BETWEEN STATIONS

The final section of the book presents a comprehensive treatment of the problems faced in interpolation from the known points of weather station measurements to specific locations where cooling installations are to be made.

DISCUSSION

H. C. S. THOM, Washington, D. C. (Written): Mr. Crow is to be congratulated on processing the largest amount of wet-bulb data so far attempted. Indeed, "Evaluated Weather Data for Cooling Equipment Design" gives an impressive amount of data which will no doubt be very valuable to engineers. In the modern sense, however, it does not give the information the engineer needs for design, for the data give no clue as to the probabilities of exceeding design limits which are essential to any design data.

The wet-bulb data Mr. Crow discusses were obtained in exactly the same way as the old TAC data which were discontinued in the GUIDE because they had no rational relation to the design values in common use. These data give the average frequency of occurrence of wet-bulb values at certain specified percentages. The percentages are not related to design probabilities in a way which can be inferred from the tables. For example, the 1% value given by the table for Fort Worth is 77.7 F and the "design value in common use" is 78 as given by the GUIDE. 1% is 0.01 probability as commonly interpreted or a mean recurrence interval of 100 years. No engineer would design for a 0.01 probability and, of course, he has not been doing this at Fort Worth because the 0.01 is not a design probability but an average frequency. Since the TAC type data do not give design probabilities, it is clear that they could not be used if design practice did not already exist so that a mean frequency, say 1%, could be chosen which would give a design value close to the one "in common use". Lacking a design probability the engineer has no rational basis for design since every design must be based on a rational measure of the risk of exceeding the design value.

The TAC type design data do not give rational design probabilities for two reasons: (a) They are computed using many hours of low load conditions when systems are either not operating or are operating at loads which are far below the design load. This results in a reduction of average frequencies to values far below rational design probabilities. (b) Since the TAC type design values are based on average frequencies they give no indication of design stresses in individual years. No engineer would think of designing on the basis of average conditions for these will be exceeded in about half of the cooling seasons. Based on average frequencies, such as the author gives, the engineer could say to a customer, "Your system will not meet the load on the average 5% of the operating hours". To be sure, over a long period the average would be 5%, but during individual years the percentage of hours of overloading must go much higher. There would be years when the system would not meet the load day after day due to the well known fact that hot weather occurs in spells. The answers to these difficulties is to compute design values so that they rationally measure the probability of failure while at the same time measuring and limiting the individual season risk. Such a system was proposed by the writer at the Dallas, Tex. meeting.

The TAC type data will be found most useful in operating problems where averages are needed. Cooling tower designers must certainly be considering other information such as "common use values" which have implied design probabilities based on experience, for the TAC data do not give the risk of exceeding design values.

AUTHOR CROW: It is completely erroneous to assume that the 1% level of summer wet-bulb data at Fort Worth published in "Evaluated Weather Data for Cooling Equipment Design" could be used to represent a probability of a recurrence interval of once in 100 years.

It is interesting that practically nothing in Mr. Thom's discussion specifically refers to material in the paper presented.

1743

No. 1743

Reaction of Refrigerant 12 with Petroleum Oils

H. O. SPAUSCHUS
Member ASHRAE

G. C. DODERER

The maximum operating temperature of a refrigeration compressor is often limited by chemical instability of the working fluid used in the systems: This fluid is a solution or mixture of refrigerant and lubricant, and the present discussion is limited to combinations of dichlorodifluoromethane (Refrigerant 12) and petroleum oils. When tested independently, both refrigerant and oil are stable for the operating life of the equipment at temperatures well above those normally attained in refrigeration compressors.[1,2] However, it has been known for some years that Refrigerant 12 and petroleum oils will react at moderate temperatures, and that the rate of reaction is influenced by the nature of the oil and accelerators, such as steel, iron oxide, moisture, etc.

As many of the studies reported are of a comparative nature, the mechanism of the reaction between Refrigerant 12 and petroleum oil has not been defined clearly. Shaw and Brandon[3] heated Refrigerant 12 and two different oils in glass-lined steel bombs, observing that one solution darkened while the other showed no change in color. No chemical analyses were performed, nor did the authors discuss the refrigerant-oil reaction. Steinle[4] used a sealed tube test developed by Philipp and Tiffany[5] to compare the reactivity of a series of oils with Refrigerant 12. His results showed that the amount of darkening of the oil-refrigerant solution was related to the "resin" content of the oil. HCl and water were formed as products of the reaction and he suggested that tar or coke is formed as polymerization product of the oil. He pointed out, however, that his studies did not

H. O. Spauschus is Research Associate and G. C. Doderer is Physicist at General Electric Co.

define the mechanism of the reaction.

Elsey, Flowers and Kelley[6] also used color change as a measure of the extent of the refrigerant-oil reaction. The general mechanism they proposed is that chlorine, and, to some extent, fluorine are split from the refrigerant molecule to combine with hydrogen atoms coming from the hydrocarbon molecules comprising the oil. These authors also analyzed the contents of several tubes for chloride, showing that chloride formation increased with time. Williamitis[7] pointed out that the "relative Refrigerant 12 deterioration" showed large variations depending on the nature of the lubricant used. He did not state how deterioration was measured. Kvalnes and Parmelee[8] studied effects of type of oil, oil viscosity, oil additives, metals, moisture, antifreeze agents and cellulose on the refrigerant-oil reaction. Their results are reported as percent refrigerant decomposed, based on chloride analysis. The authors assumed stoichiometric decomposition of Refrigerants 12 and 22 to chloride.

TABLE I

PHYSICAL PROPERTIES AND CARBON-TYPE OF PETROLEUM OILS

	Oil A		Oil B		
Density, 20 C	0.8823		0.9149		
Refractive index, N_D^{20}	1.4780		1.4980		
Viscosity, cs at 20 C	108		92		
ssu at 100 F	192		156		
Molecular weight (Cryoscopic)	314		299		
Carbon Type	$\%C_P$	$\%C_N$	$\%C_P$	$\%C_N$	$\%C_A$
n-d-M[10]	45	55	35	57	8
VGC - Ri[11]	45	55	36	56	8
Mean	45	55	35.5	56.5	8

Walker, Rosen and Levy[9] have reported effects of water, air, temperature, time and various metal combinations on mixtures of Refrigerant 22 and three refrigerating oils. Tube contents were rated for color, viscosity change, sediment, wall deposits and copper plating as measures of the extent of reaction. No attempt was made to elucidate the course of the reactions or to identify reaction products.

The purpose of the present study was to gain a clearer understanding of the Refrigerant 12-petroleum oil reaction rather than to obtain more comparative data. A careful study of the reaction products formed at different temperatures and times provides information about the mechanical and kinetics of the reactions.

Materials and Experimental Procedures—Refrigerant 12 from a single cylinder was used for calibration of the mass spectrometer and for sealed tube experiments. Two different petroleum oils were studied: Oil A, a highly refined "water-white" oil, and Oil B, a medium refined naphthenic oil recovered from Gulf Coast crudes. Physical properties and carbon-type composition of each are given in Table I.

Known quantities of Refrigerant 12 and oil were sealed into glass combustion tubes (200 mm length, 8 mm ID, 10 mm OD) which were then heated, and the contents analyzed quantitatively.

Details of the method of preparing the sealed tubes have been presented previously.[8, 9, 12] As well as charging in refrigerant and oil, a small strip of compressor valve steel weighing 0.048 gm was added to each tube prior to sealing. Precautions were taken to eliminate moisture and air from the oil and to control other variables, such as tube size and metal surface conditions. Instead of being placed directly in an oven, the sealed tubes were inserted into a stainless steel cylinder, which was partially filled with liquid Refrigerant 12 and then closed. This produced an excess pressure on the outside of the glass tubes, since the saturation pressure of pure Refrigerant 12 is greater than that of refrigerant-oil solutions. Consequently, the pressure differential between inside and outside the glass tubes was reduced greatly and glass tube breakage in the oven was eliminated.

After the heating period, the Refrigerant 12 was discharged from the stainless steel cylinder, the tubes removed and prepared for analysis. A sample tube was partially immersed in liquid nitrogen to freeze the refrigerant and lower internal pressure, reducing the possibility of premature breakage when a scratch was filed near the tip. The tip of the cold tube was inserted into the sampling fixture shown in Fig. 1 through a snugly fitting vacuum hose section, the tube being pushed in until the file scratch was even with the end of the copper tubing. Liquid nitrogen was again applied, and the sampling fixture was connected to the mass spectrometer through a calibrated gas handling system, as shown in Fig. 2. The gas system and sampling fixture were evacuated, until the thermocouple vacuum gauge indicated a pressure of less than ten microns Hg. The evacuation port was closed, liquid nitrogen removed, the sample tube

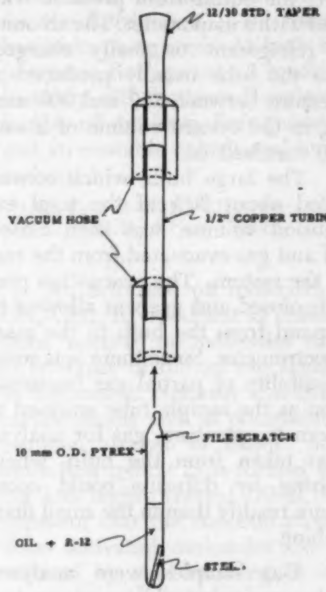
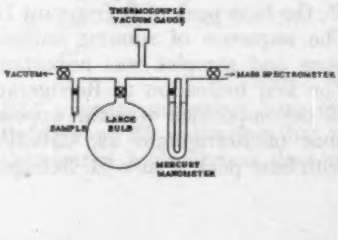


Fig. 1 Sampling fixture

Fig. 2 Calibrated gas handling system



grasped with a gloved hand, and sideways pressure exerted to break off the glass tip against the end of the copper tubing. Liquid nitrogen noncondensable gases expanded immediately. As the tube warmed, the refrigerant and other condensed gases gradually volatilized into the known volume system. After the tube had warmed to room temperature, the equilibrium pressure was read with a manometer. The amount of refrigerant originally charged into the tube usually produced a pressure between 250 and 300 mm Hg in the system volume of about 800 standard cc.

The large bulb, which constituted about 80% of the total expansion volume, was then closed off and gas evacuated from the rest of the system. The evacuation port was closed and gas was allowed to expand from the bulb to the mass spectrometer. Since there was some possibility of partial gas fractionation as the sample tube warmed to room temperature, gas for analysis was taken from the bulb, where mixing by diffusion could occur more readily than in the small diam tubing.

Gas samples were analyzed using an analytical mass spectrometer. After manually recording hydrogen, the instrument was set to automatically scan upward from m/e 12, usually stopping at m/e 85, the base peak of Refrigerant 12. The sequence of running calibrations and samples was important. The first indication of Refrigerant 12 decomposition was the appearance of Refrigerant 22, $CHClF_2$, with base peak at m/e 51. Refrigerant 12 contributed less than 0.20%, relative to its base peak, to the 51 peak. Mass spectrometer cracking patterns of a gas normally show slight variations from one run to the next. Greatest variance found in the 51 peak during Refrigerant 12 calibrations indicated that Refrigerant 22 could be detected with certainty, when 0.10 standard cc or more was present in the entire gas sample from the tube.

The base peak of hydrogen chloride is at m/e 36. Refrigerant 12 contributed about 1.8% of its base peak to the 36 peak. A similar check of calibration pattern variations indicated 0.20 standard cc to be the detection limit for HCl.

Refrigerant 12 was run first to record its cracking pattern and establish its contributions at m/e 36 and 51. Experimental samples were run next. In order to realize the greatest sensitivity for Refrigerant 12 decomposition with the least interference from spectrometer background, the tubes for a day's analysis were arranged according to degree of fluid darkening and amount of sludge accumulation. Samples having little or no decomposition were run first while the spectrometer tube had negligible background. Tubes with the most decomposition were analyzed last, and scans to determine HCl background corrections were often necessary between sample runs.

After all sample patterns were recorded and another background determined, Refrigerants 12 and 22 and HCl were calibrated in that order. Lastly, calibrations were performed for hydrogen and other

gases detected in the samples. Using relative sensitivities determined from the gas calibrations and appropriate corrections for coincident peaks, percentage compositions of the gas samples were calculated. These percentages were then used with the measured volume of gas sample to calculate the amount of each component present, expressed as standard cc of gas.

RESULTS AND DISCUSSION

Mechanism of Refrigerant 12-Oil Reaction — Series of experiments were conducted at 175 and 200 C with tubes heated for periods of 7, 14 and 28 days. Results are summarized in Table II. In these experiments the ratio of weight refrigerant to weight oil was held nearly constant in the range 1.7 to 1.9. All results are the average of duplicate or triplicate experiments.

TABLE II

STANDARD CC OF GASES FORMED BY REFRIGERANT 12-OIL REACTION

	175 C					
Oil	A			B		
Time (days)	7	14	28	7	14	28
HCl	—	—	—	3.4	12.2	55
CO_2	—	—	—	0.09	0.36	3
Refrigerant 22	0.07	0.10	0.23	2.3	7.2	24
	200 C					
Time (days)	7	14		7	14	
CH_4	—	—		—	0.3	
H_2	—	—		1.7	5.0	
CO	—	—		—	1.3	
HCl	0.3	0.9		69.	179.	
CO_2	—	—		6.2	25.	
Refrigerant 22	1.1	2.1		34.	59.	

Blanks in the table indicate absence of the gas.

The first gas to appear, and hence, the most sensitive indicator for the reactions, was Refrigerant 22. As conditions became more severe (higher temperature, longer reaction time, less highly refined oil), new reaction products appeared in the order HCl, CO_2, CO, and CH_4. From these results, a mechanism for the Refrigerant 12-oil reaction has been postulated. At high temperatures Refrigerant 12 reacts selectively with "activated" molecules in the oil to form Refrigerant 22 and an unstable chlorinated hydrocarbon.

$$CCl_2F_2 + R\text{-}CH_2\text{-}CH_3 \longrightarrow \underset{\displaystyle Cl}{\underset{|}{R CH}}\text{-}CH_3 + CHClF_2 \qquad (1)$$

An activated oil molecule, R-CH_2-CH_3, is visualized as differing from "normal" oil molecules either in structure of the carbon skeleton or because it contains non-hydrocarbon atoms such as phosphorous, nitrogen, oxygen or sulfur. Since more Refrigerant 22 is formed with the less highly refined Oil B, it is apparent that the concentration of these activated molecules can be reduced by acid washing or clay adsorption refining processes.

At high temperatures the chlorinated hydrocarbon dehydrohalogenates to produce HCl and an unsaturated hydrocarbon.

$$R\text{-}\underset{\displaystyle Cl}{\underset{|}{CH}}\text{-}CH_3 \longrightarrow RCH = CH_2 + HCl \qquad (2)$$

HCl appears somewhat later than Refrigerant 22, indicating that the chlorinated hydrocarbon does not

decompose immediately. Experimental data show that the concentration of halogenated oil is usually small, since approximate chloride balance exists between initial Refrigerant 12 and final 12 unreacted + 22 + HCl. For example, in a typical experiment the tube was charged with 1.485 gm of Refrigerant 12. After heating, the volumes of gases found were 160 cc of Refrigerant 12, 59 of Refrigerant 22 and 166 of HCl. The chloride balance is:

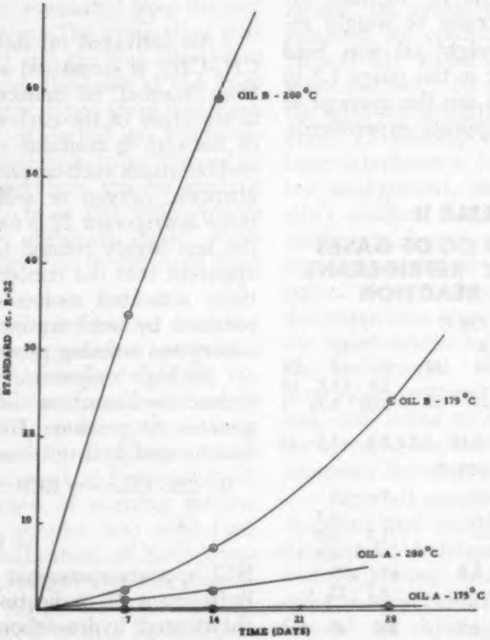
gm Cl from HCl	0.263
gm Cl from Refrigerant 22	0.094
gm Cl from Refrigerant 12	0.507
Total	0.864
gm Cl from original Refrigerant 12	0.870

When the concentration of unsaturated hydrocarbon molecules becomes sufficiently large, they polymerize to form tar or coke.

$$n\,RCH=CH_2 \longrightarrow (\underset{|}{RCH}-\underset{|}{CH_2})_n \quad (3)$$

HCl, as formed by reaction (2) can attack iron or steel present in the system, forming hydrogen and iron chloride.

Fig. 3 Rate of Refrigerant 22 formation



$$2\,HCl + Fe \longrightarrow H_2 + FeCl_2$$
or
$$6\,HCl + 2Fe \longrightarrow 3H_2 + 2FeCl_3 \qquad (4)$$

Quantities of hydrogen are small, indicating this reaction is of minor magnitude for conditions studied. Microscopic examination of the steel strips, after high temperature tests, revealed a thin surface film in all cases where hydrogen was formed. Unfortunately, the film was too thin to confirm the presence of iron chlorides by X-ray diffraction analysis.

Previous studies of the thermal decomposition of Refrigerants 12 and 22 provide clues to the mechanism of CO_2 and CO formation observed for the more severe conditions. Norton[2] has shown that Refrigerant 12 is relatively stable up to temperatures as high as 500 C. In the presence of catalysts the decomposition rate is enhanced, but even with Fe_2O_3, the most effective catalyst, the decomposition rate is too low to account for the CO_2 found in our experiments. Refrigerant 22, however, decomposes at much lower temperatures with the formation of CO and CO_2. In the presence of effective catalysts, 1% of the Refrigerant 22 is decomposed per hr at 150 C. The overall reaction for experiments carried out in glass tubes is given by Norton as

$$CHClF_2 \xrightarrow[SiO_2]{Fe} CO_2 + CO + SiF_4 + 2HCl \qquad (5)$$

In our experiments, the partial pressure of Refrigerant 22 was relatively low and the iron surface covered with oil-refrigerant solution, thus retarding the rate of Refrigerant 22 thermal decomposition. In keeping with this hypothesis, our results reveal increased CO_2 formation with increasing temperature, time and Refrigerant 22 concentration. The increase in HCl/$CHClF_2$ ratio under severe conditions is further support for reaction (5).

Trace quantities of methane were detected under the most extreme reaction conditions studied, probably being produced by thermal cracking of the oil.

$$R - CH_2 - CH_3 \longrightarrow R' - CH_3 + CH_4 \qquad (6)$$

In the postulated mechanism, Refrigerant 22 is formed by reaction (1) and some of it subsequently decomposes to form CO_2 and CO by reaction (5). Other studies[8,9] have shown that Refrigerant 22 will also react directly with oil, although not as readily as Refrigerant 12. It has not been possible to obtain information about this reaction in the present series of tests because of

TABLE III
EFFECT OF REFRIGERANT 12-OIL RATIO ON GASEOUS REACTION PRODUCTS AT 200 C

14 Days

Oil	A			B		
Refrigerant 12/Oil	1.16	1.70	3.13	1.02	1.88	3.70
CH_4	—	—	—	0.3	0.9	+
H_2	—	—	—	2.0	3.6	3.6
CO	—	—	—	1.4	2.5	1.7
HCl	—	—	0.15	151	164	159
CO_2	—	—	—	20	23	21
Refrigerant 22	0.42	0.90	2.3	48	55	59

the complexity of the reactions preliminary to or concurrent with the Refrigerant 22 reaction. Substituting Refrigerant 22 for 12 in the sealed tubes would be a more direct and simpler method of studying this reaction.

Our experiments have established that large quantities of Refrigerant 22 are formed, thus proving that the decomposition of 12 is not stoichiometric with respect to chloride formation as suggested by Kvalnes and Parmelee. Although the reactions involved are complex, the quantity of chloride will usually provide a relative measure of the Refrigerant 12 decomposition rate.

Effect of Refrigerant 12-Oil Ratio on the Reaction – A series of tests was conducted to see what effect the ratio of Refrigerant 12 to oil had on the type and quantity of reaction products. Results for both oils at 200 C, 14 days, are given in Table III.

For the combination Refrigerant 12-Oil B, which produces large quantities of reaction products, there are no significant changes in the type or quantity of gases formed in increasing the Refrigerant 12 to oil ratio by a factor of more than three. Variances in quantity of the individual gases are within the magnitude of error expected in these measurements. A possible exception is the small but regular increase in Refrigerant 22. These results imply that reaction (1) approaches zero order for oils containing a high concentration of "activated" molecules. The rate of reaction may be limited by the rate of diffusion of reactants or products through the liquid phase.

For Oil A, which produces small quantities of reaction products, Refrigerant 22 formation increases rapidly with increase in the Refrigerant 12 to oil ratio. An approximation calculation indicates the reaction for this combination is first order with respect to Refrigerant 12 concentration. The weight of Refrigerant 12 charged into the tubes provides a measure of the initial concentration, a, and it is assumed that one molecule of Refrigerant 12 has reacted for each molecule of 22 detected. The quantity reacted is designated as x. For a first order reaction

TABLE IV
REACTION RATE CALCULATIONS, 200 C

	Initial Concentration gm Refrigerant 12	standard cc Refrigerant 12 (a)	Quantity Reacted (x)	$\frac{a}{a-x}$
Oil A	0.958	177.5	0.42	1.002
	1.587	294.1	0.90	1.003
	2.965	549.4	2.3	1.004
Oil B	0.962	178.3	48	1.37
	1.580	292.8	55	1.23
	3.247	601.7	59	1.11

$$k = \frac{2.303}{t} \log \frac{a}{a-x}$$

and at constant time

$$\log \frac{a}{a-x} = \frac{kt}{2.303} = K$$

Results are summarized in Table IV.

The ratio $\frac{a}{a-x}$ is nearly constant for Refrigerant 12 with Oil A indicating a first order reaction as a distinct possibility. For Oil B, the ratio $\frac{a}{a-x}$ decreases with increasing Refrigerant 12 concentration.

Rates of Reaction – Complexity of the Refrigerant 12-oil reactions makes it difficult, if not impossible, to interpret the results rigorously in terms of chemical kinetics. The system is heterogeneous and liquid phase concentrations at high temperatures can only be approximated. Presence of steel catalyzes the reactions. Some reaction products themselves are not stable and undergo further reactions.

Data for Oil B at 175 C, as given in Table II, were fitted to the first order rate equation using Refrigerant 22 quantities as a measure of the amount of Refrigerant 12 decomposed. The plot of log fraction reacted versus time was not linear. No calculations were made for Oil A-Refrigerant 12 reactions because the quantities of Refrigerant 22 found were extremely low, and small errors would be magnified many times in rate calculations.

In discussing the reaction mechanism, it was noted that Refrigerant 22 is the most sensitive indicator of Refrigerant 12 reaction. Thus, the Refrigerant 22 content, at any given time, provides a relative measure of the extent of the reaction. In Fig. 3, the amounts of Refrigerant 22 are plotted as a function of time for both oils at 175 and 200 C. Although some of the Refrigerant 22 undergoes further reactions, the curves in Fig. 3 provide a means of comparing the high temperature reactivity of different oils with Refrigerant 12 and illustrate important influences of the type of oil and temperature on the reactions. Oil A with Refrigerant 12 at 200 C is considerably more stable than Oil B with Refrigerant 12 at 175 C.

The high sensitivity of mass spectrometer analysis provides an advantage to this method as compared with previously published studies. In a period of days, differences in reactivity of various oils are readily distinguishable in tests at 175 C. At higher temperatures this time can be reduced even further. However, the probability of complicating side reactions places a definite upper limit on the temperature at which realistic tests can be conducted.

REFERENCES

1. Physical Chemistry of Lubricating Oils, A. Bondi (Reinhold Publishing Corporation, New York, 1951, p. 329).

2. Rates of Thermal Decomposition, F. J. Norton (Refrigerating Engineering, Vol. 65, No. 9, September 1957, p. 33).
3. A. H. Shaw and A. Brandon (Proceedings of the Institute of Refrigeration, Vol. 44, 1947-8, p. 95).
4. H. Steinle (Kaltetechnik, Vol. 2, No. 7, 1950, p. 174).
5. Thermodynamics of Sulfur Dioxide-Oil Systems, L. A. Philipp and B. E. Tiffany (Refrigerating Engineering, Vol. 27, No. 5, May 1934, p. 248).
6. A Method of Evaluating Refrigerator Oils, H. M. Elsey, L. C. Flowers and J. B. Kelley (Refrigerating Engineering, Vol. 60, No. 7, July 1952, p. 737).
7. V. A. Williamitis (General Motors Engineering Journal, Vol. 2, No. 1, 1955, p. 22).
8. Behavior of Refrigerants 12 and 22 in Sealed Tubes, D. E. Kvalnes and H. M. Parmelee (Refrigerating Engineering, Vol. 65, No. 11, November 1957, p. 40).
9. A Study of the Factors Influencing the Stability of the Mixtures of Refrigerant 22 and Refrigerating Oils, W. O. Walker, S. Rosen and S. L. Levy (Presented at the ASHRAE 67th Annual Meeting in Vancouver, B. C., June 13-15, 1960).
10. Aspects of the Constitution of Mineral Oils, K. Van Nes and H. A. Van Westen (Elsevier Publishing Company, Inc., New York, 1951, p. 340).
11. S. S. Kurtz, R. W. King, W. J. Stout, D. G. Partikian and E. A. Skrabek (Analytical Chemistry, Vol 28, 1956, p. 1928).
12. Kinetic Technical Bulletin 14 (Freon Products Div., E. I. DuPont de Nemours & Company, Inc., 1953 and 1955.

DISCUSSION

I. A. Naman, Houston, Tex.: It is my understanding that in addition to the production of hydrochloric acid and other contaminants, some soaps are also produced. These soaps are able to take up copper in the cooler parts of the system and then throw out this copper in some way in the hotter parts, such as the condenser, where the oil containing the copper gets into the condenser and deposits this copper out in little particles into the bearings and system.

How was it determined whether there were nonvolatile contaminants present or produced during the heating process, which were not discovered through the technique described?

Author Spauschus: A technique was developed whereby the sludge is placed in a vacuum furnace which leads directly to the mass spectrometer. The temperature is raised at 50 C intervals until the sludge begins to volatilize. This results in certain patterns characteristic of fragments which come off at various levels of temperature. Some of these are hydrocarbons, chlorides, etc.; the molecular structure is not known specifically, but is characteristic of the sludge. Using this technique, we have established that the black residue formed in sealed tube tests is identical to deposits removed from compressors operated at excessive temperatures.

W. O. Walker, Coral Gables, Fla.: Since color was used as a rough mechanism for beginning the determinations, was there any specific correlation between the color in some of the less active tubes and the findings of the percentage of Refrigerant 22?

Author Spauschus: The correlation is quite good, especially in those cases where reaction has proceeded to quite an extent. When a brownish color appears and two tubes are placed adjacent to one another, it is possible to tell which one has reacted further. However, at the lower levels of decomposition there may or may not be a color change, and, therefore, there may be some uncertainty. In such cases, the quantitative data are superior.

1744

No. 1744

Sizing of Refrigeration System Pipelines for Optimum Economy

DONALD J. RENWICK

Most methods of pipe sizing for refrigeration systems have been based on allowable pressure drop or else a suitable range of velocity of flow, where it was assumed that the recommended values to be used were determined by investigators so as to result in reasonable economy and operating conditions; reasonable insofar as the annual investment cost of the piping as installed, together with the pumping cost to overcome friction losses of the fluid flowing, would be near a minimum total cost of owning and operating the system.

Since the cost of electricity per kw hr has been gradually decreasing as time progresses, while the manufacturing and raw material costs for piping as well as installing labor costs have all risen considerably in recent years, a review of the economics of pipe size selection for electric motorized refrigeration systems is thought to be opportune.

As a consequence, a series of pipe selection charts, which are based on a combination of analytical and graphical methods of analysis of the economic parameters affecting the optimum sizing of refrigeration pipelines, has been devised and they are herein presented. It is felt the charts are both flexible in that they allow for changing economic conditions as well as being complete in their inclusion of the major factors that should be considered in the selection of pipe sizes for optimum economy.

REVIEW OF ECONOMIC ITEMS INVOLVED

Fig. 1 illustrates schematically the economic pattern involved in pipe size selection. Curve "A" shows how the cost of overcoming friction will decrease as the diameter of pipe is selected toward larger sizes.

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This pumping cost will accrue from the extra work of running the compressor in order to maintain fluid flow through the pipe lines and can be evaluated in terms of added cost of electricity for any electric motor driven compression type of system.

Curve "B" represents the typical trend of installed pipe cost, which naturally increases as larger diameters are selected. The total cost, represented by curve "C," which is obtained by adding curves A and B together, illustrates that a definite minimum cost or optimum diameter will occur. Several parameters influence what this optimum diameter should be in any given design and a mathematical-graphical method will now be developed for determining the diameter of both suction and discharge lines of refrigeration systems for several popular refrigerants. (See Appendix for construction details of Fig. 1.)

FUNDAMENTAL EQUATIONS AND NOMENCLATURE

The fundamental analysis and development of a method for pipe sizing proceeds as follows:

Total annual cost = cost of fluid friction per year + annual investment cost of pipe as installed

Work of fluid friction = $\Delta P\ v\ w$ (in ft-lb per sec)

Cost of fluid friction per year =

$$\frac{8760\ \Delta P\ v\ w\ U_f\ E}{738\ \eta_e} \text{ (in ¢/year)}$$

$$\Delta P = \frac{f\ L_e\ \rho\ V^2}{2gD} = \text{Fanning equation}$$

for friction loss in lbs per ft^2

$$f = \frac{0.184}{(R_e)^{.2}} = \text{Pipe friction factor}$$

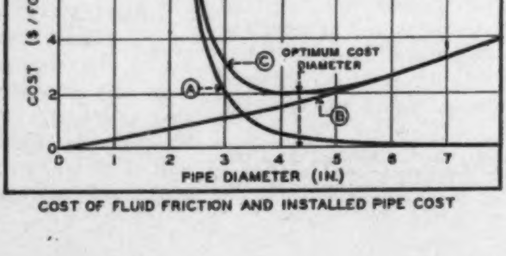
for copper tubing

$$f = \frac{0.23}{(R_e)^{.2}} = \text{Pipe friction factor for}$$

steel tubing (estimate at 25% higher than for copper)

$$R_e = \frac{d V \rho}{.00806\ \mu} = \text{Reynolds number}$$

Fig. 1



COST OF FLUID FRICTION AND INSTALLED PIPE COST

d = Inside pipe diam in., D = diam ft
L_e = Equivalent length of piping system, ft
L = Actual length of piping system, ft
ρ = Density of fluid flowing in lb per cu ft
R.E. = Refrigeration effect in Btu lb
T = System capacity in ton

$$V = \frac{576\ w\ v}{\pi\ d^2} = \text{Fluid velocity in fps}$$

v = Average specific volume of fluid in cu ft per lb

$$w = \frac{T \times 200}{60 \times R.E.} = \text{lb of fluid flowing per sec}$$

E = Electrical cost in cents per kw hr
η_e = Overall isentropic compression and motor efficiency combined (%/100)
U_f = Usage factor as fraction of year running time (%/100)
μ = Fluid viscosity in centipoises
8760 = hr per year
738 = Conversion factor for ft-lb per sec to kw

Making all the appropriate substitutions as indicated above and combining constants, the cost equation then becomes:

Total annual cost =

$$\frac{5.37 \times 10^4\ \mu^{.2}\ v^2\ T^{2.8}\ E\ L_e\ U_f}{(RE)^{2.8}\ \eta_e\ d^{4.8}} + L \times$$

Graph of installed pipe cost—¢ per ft per yr basis

Now to determine the optimum diameter the above equation should be differentiated with respect to diameter and the result set equal to zero, thus:

$$\frac{\delta}{\delta d}\text{(Total Cost)} = 0 = \frac{2.58 \times 10^5\ \mu^{.2}\ v^2\ T^{2.8}\ E\ \dfrac{L_e}{L}\ \dfrac{1}{\eta_e}\ U_f}{(R.E.)^{2.8}\quad d^{5.8}} +$$

Graphically differentiated cost of pipe per foot

which in effect indicates that the diameter will be optimum when the differentiated value of the friction cost term exactly equals the differentiated term for installed pipe cost.

This last term of the equation (for installed pipe cost) is more conveniently handled graphically and is so indicated above, since there is no universally simple equation or expression for the installed cost of piping based on variation with diameter. Several references (notably 3, 4, and 5) show graphical presentations of pipe cost and clearly illustrate the natural rising trend of cost with increasing diameter. The total installed cost of pipe (on a per ft basis) is of course higher than the purchase price of straight pipe alone, since the cost of fittings and installing labor must be included in the total.

However, the proportional trend is still maintained and some investigators (4 and 6) have noted the average installed cost of pipe increases to about the 1.5 power of diameter. However, this power factor varies from less than 1 to greater than 2 depending on such items as type of pipe, number of fittings, individual manufacturers' current pricing schedule, quantity discounts, shipping costs, and installing labor wage scales.

Fig. 2 illustrates a typical pipe cost variation with diameter; and since this is a log log plot, it indicates (by slope of curve) a somewhat less than unity power factor for steel pipe sizes less than 1 in., and about 1:1 ratio of cost to diam in the mid-range of 1 to 3-in. pipe sizes, and for sizes larger than 3 in.

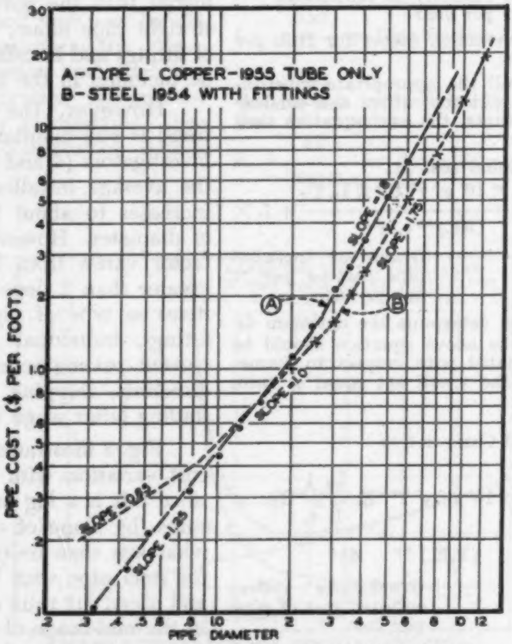
the power exponent approaches 2.

A similar plot of a typical copper tube price list indicates a slope of about 1.25 for sizes smaller than 2 in. and about 1.8 for sizes over 2 in. diam. It is assumed that regardless of all the factors which influence the final installed price of a given piping system, a reasonably accurate price estimate can be made from a cost graph based on an average system. The appendix then presents a method for differentiating such a curve and adapting it to the charts presented here for optimum pipe size selection.

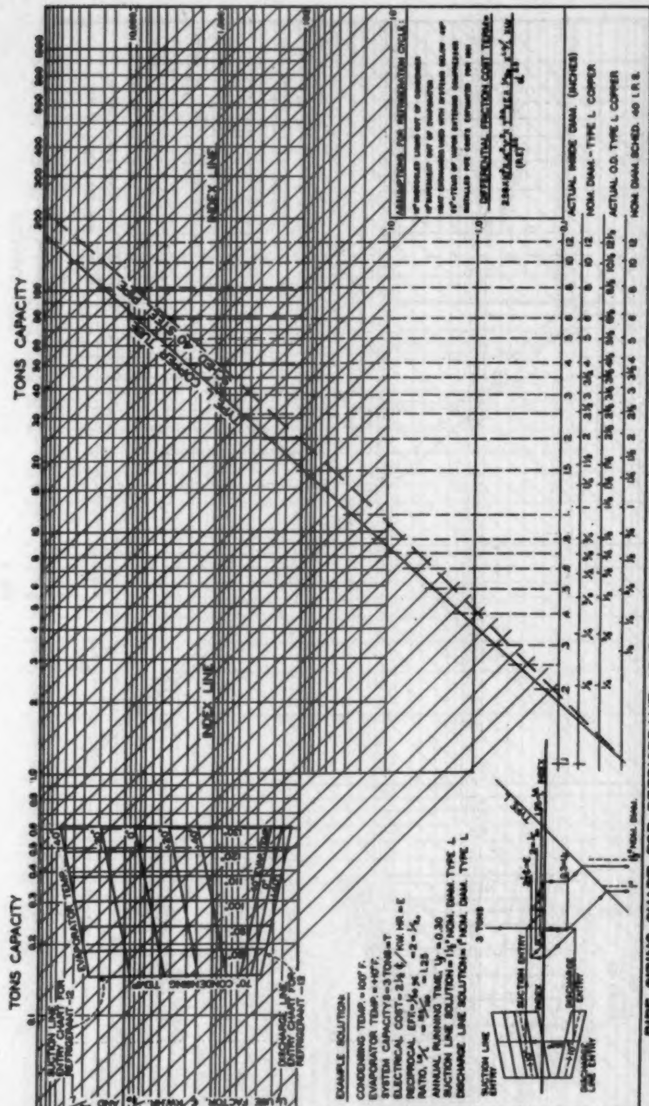
CONSTRUCTION AND USE OF PIPE SELECTION CHARTS

Figures 3, 4, and 5 illustrate pipe size selection charts for Refrigerants 12 and 22 and ammonia respectively. The sloping lines labelled "copper-type L" and "steel-sched. 40" represent the effect of installed cost of either copper or steel piping and are located (as referred to above) by the method outlined in the appendix. These pipe cost lines may occasionally need to be relocated, but only at times of appreciable pipe cost changes.

Fig. 2

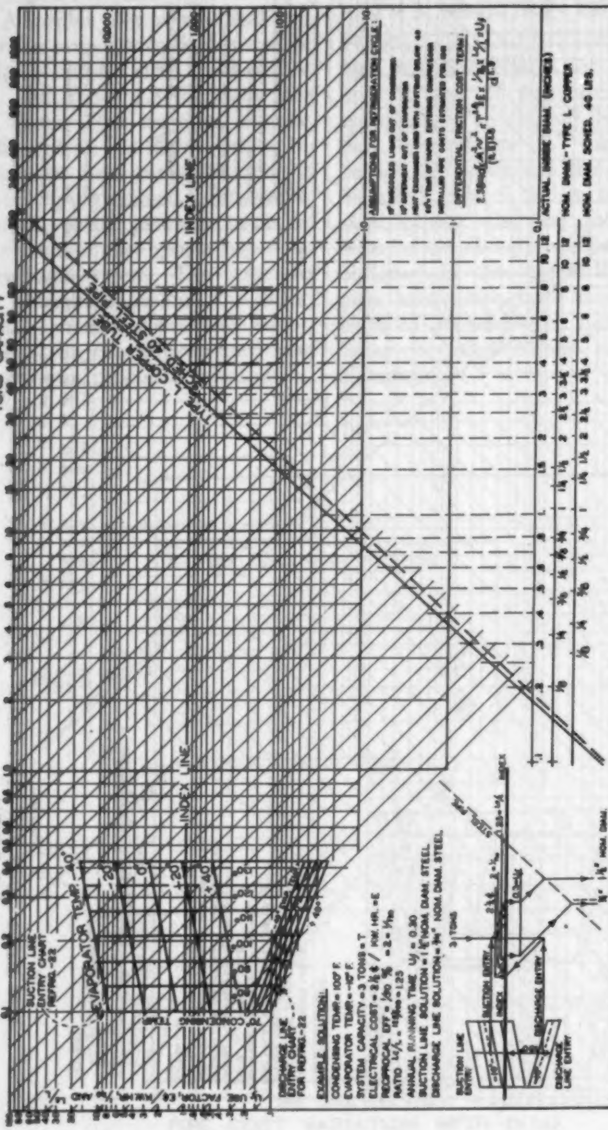


PIPE COST VARIATION WITH DIAM.



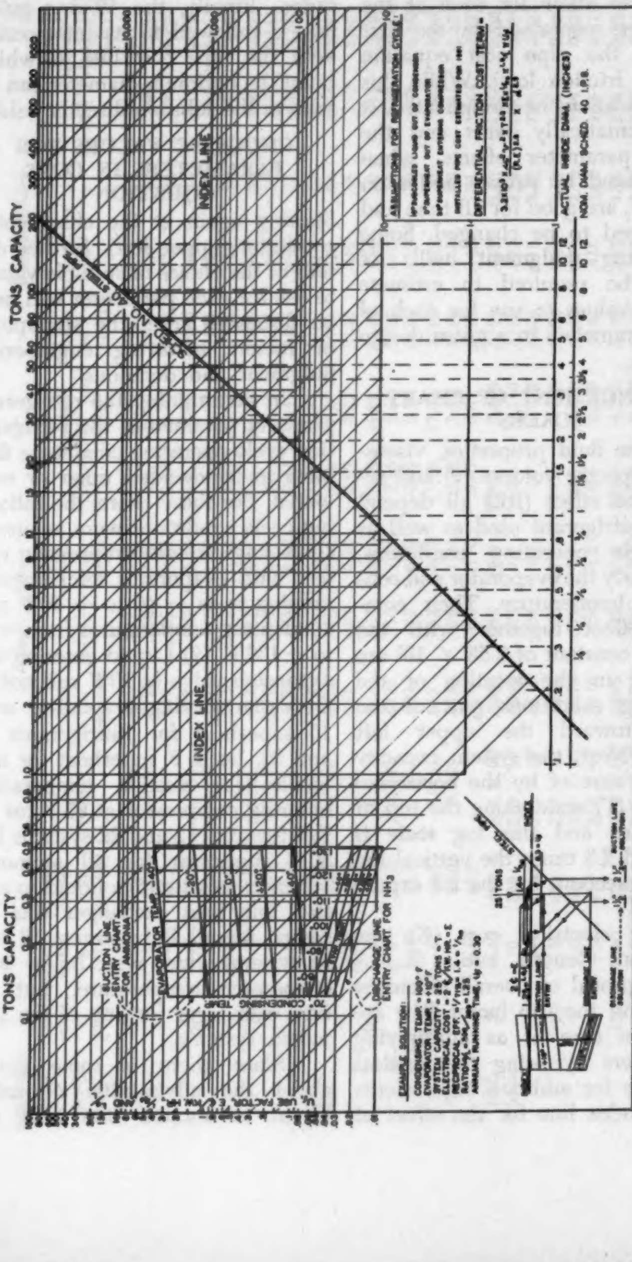
PIPE SIZING CHART FOR REFRIGERANT-12 SUCTION AND DISCHARGE LINES FOR OPTIMUM ECONOMY

Fig. 3



PIPE SIZING CHART FOR REFRIG-22 SUCTION AND DISCHARGE LINES FOR OPTIMUM ECONOMY

Fig. 4



PIPE SIZING CHART FOR AMMONIA SUCTION AND DISCHARGE LINES FOR OPTIMUM ECONOMY

Fig. 5

The rest of the chart is constructed to allow for each of the parameters contained in the first term of the pipe cost equation (i.e., for friction loss). While this term appears to be complicated, it is mathematically exact and the several parameter effects, when once located by proper scales on the chart, are good for all time and never need to be changed. Some engineering judgment will, of course, be required to estimate suitable values to use for each of these parameters in a given design situation.

ARRANGEMENT OF CHART SCALES

The three fluid properties, viscosity (μ), specific volume (v) and refrigeration effect (RE) all depend on the refrigerant used as well as the cycle operating conditions, particularly the evaporator and condensing temperature. Their combined effect together with the formula constant of 2.58×10^5 are included in the location of the operating conditions part of the chart (toward the upper left corner). Next, the system capacity is taken care of by the horizontal tonnage (T) scale along the top of the chart, and this log scale is stretched 2.8 times the vertical log scale to account for the 2.8 exponent of T.

The electrical cost (E), the equivalent length ratio (L_e/L), the reciprocal of overall efficiency ($1/\eta_o$) and the use factor (U_f) are all taken care of as multiplying parameters by using the vertical log scale for addition adjustments at the index line for the effect of each parameter taken in successive order. Finally, the 45 deg guide line is followed to its intersection with the pipe cost line at which point the optimum diameter can be read as the solution directly below.

CHOOSING VALUES FOR PARAMETERS IN COST EQUATION

Starting again at the chart entry condition (upper left), further details of the choice of suitable values to use for each parameter will now be discussed from the standpoint of factors involving engineering judgment and decisions.

In determining the refrigerant property parameters, the refrigeration cycle operating conditions first have to be decided upon or estimated. For these charts the following cycle conditions were assumed: (1) A single stage compression system with evaporator temperatures ranging from − 40 to + 40 F and condensing temperatures from +70 to +130 F; (2) Liquid entering the expansion valve at 10 F subcooled; (3) Vapor leaving evaporator with 10 superheat for Refrigerants 12 and 22, but 5 F superheat for ammonia; (4) Suction temperature entering compressor at 65 F for all Refrigerants 12 and 22 systems but 30 F superheat for all ammonia systems, because (5) a liquid to suction line heat exchanger was assumed for all Refrigerants 12 and 22 systems which were below 40 F evaporator temperature, but no heat exchanger for any of the ammonia systems.

Since there is some doubt about the advisability of using liquid to suction line heat ex-

TABLE I — VALUES OF REFRIGERANT PARAMETERS FOR FRICTION COST CHART CONSTANTS

REFRIG-ERANT & PIPE MAT.	Cond. temp. t_3 (°F)	Evap. temp. t_2 (°F)	Refrig. Effect h_2-h_1 (Btu/lb)	$(R.E.)^{2.8}$	Suction Line: Visc. μ_s (cent)	Suction Line: $(\mu_s)^{.2}$	Suction Line: Spec. Vol. v_s ft³/lb	Suction Line: $(V_s)^2$	Suction Line: Chart* Constant $\frac{\phi_1(\mu_s)^{.2}(v_s)^2}{(R.E.)^{2.8}}$	Discharge Line: Visc. μ_d (cent)	Discharge Line: $(\mu_d)^{.2}$	Discharge Line: v_d	Discharge Line: $(v_d)^2$	Discharge Line: Chart* Constant $\frac{\phi_1(\mu_d)^{.2}(v_d)^2}{(R.E.)^{2.8}}$
12 Cu	70	—40	62.2	1.045	.012	.412	5.10	26.0	26.40	.0155	.435	.68	.462	0.497
		0	62.2	× 10⁵	.0125	.416	1.87	3.5	3.60	.0145	.429	.59	.348	0.354
		+40	62.2		.013	.423	0.82	0.67	0.70	.0140	.426	.53	.281	0.284
	100	—40	55.3	0.756	.012	.412	5.10	26.0	36.60	.0160	.437	.46	.212	0.316
		0	55.3	× 10⁵	.0125	.416	1.87	3.5	4.97	.0155	.435	.40	.160	0.237
		+40	55.3		.0135	.423	0.82	0.67	0.97	.0155	.435	.35	.123	0.182
	130	—40	48.0	0.508	.012	.412	5.10	26.0	54.50	.0165	.440	.31	.096	0.214
		0	48.0	× 10⁵	.0125	.416	1.87	3.5	7.40	.0165	.440	.27	.073	0.162
		+40	48.0		.0135	.423	0.82	0.67	1.44	.0165	.440	.23	.058	0.118
22 Cu	70	—40	82.5	2.32	.012	.412	4.17	17.35	7.95	.0160	.437	.63	.397	0.193
		0	82.5	× 10⁵	.0125	.416	1.60	2.56	1.29	.0150	.432	.54	.292	0.141
		+40	82.5		.013	.423	0.70	0.49	0.23	.0140	.426	.46	.212	0.101
	100	—40	73.2	1.67	.012	.412	4.17	17.35	11.06	.0175	.445	.42	.177	0.088
		0	73.2	× 10⁵	.0125	.416	1.60	2.56	1.65	.0165	.440	.36	.130	0.064
		+40	73.2		.013	.423	0.70	0.49	0.32	.0155	.435	.30	.092	0.045
	130	—40	63.0	1.086	.012	.412	4.17	17.35	17.00	.0190	.452	.30	.090	0.045
		0	63.0	× 10⁵	.0125	.416	1.60	2.56	2.56	.0180	.448	.25	.063	0.031
		+40	63.0		.013	.423	0.70	0.49	0.49	.0175	.445	.21	.044	0.022
NH_3 Steel	70	—40	490	340 × 10⁵	.0085	.385	27.0	729.0	2.660	.0152	.433	3.80	14.45	0.590
		0	505	370 × 10⁵	.0098	.397	9.9	98.0	0.340	.0134	.422	3.25	10.60	0.039
		+40	516	393 × 10⁵	.0110	.406	4.4	19.4	0.065	.0124	.416	2.80	7.84	0.027
	100	—40	456	278 × 10⁵	.0085	.385	27.0	729.0	3.250	.0170	.442	2.50	6.25	0.032
		0	471	305 × 10⁵	.0098	.397	9.9	98.0	0.410	.0153	.434	2.20	4.84	0.022
		+40	482	325 × 10⁵	.0110	.406	4.4	19.4	0.079	.0142	.426	1.90	3.61	0.015
	130	—40	421	223 × 10⁵	.0085	.385	27.0	729.0	4.050	.0190	.452	1.80	3.24	0.021
		0	436	245 × 10⁵	.0098	.397	9.9	98.0	0.510	.0170	.442	1.55	2.40	0.014
		+40	447	263 × 10⁵	.0110	.406	4.4	19.4	0.096	.0162	.438	1.30	1.69	0.009

NOTES: *$\phi_1 = 2.58 \times 10^5$ for copper lines
$\phi_1 = 3.22 \times 10^5$ for steel lines
See Fig. 8 for refrig. cycle subscripts

h_2 assumed at 84 Btu/lb for Refrigerant-12 at all evap. temps. and h_2 = 110.5 Btu/lb for Refrigerant-22 because of heat interchanger effect. h_1 taken for 10 F subcooling of liquid.

changers in Refrigerant 22 systems (see Reference 2, chap. 53), the entry constant was calculated for cycles both with and without the heat exchanger present; however, the difference is negligible, so the chart entry constants are based on cycles with the heat exchanger included. Table I lists some of the values used over the range of evaporator and condensing temperature conditions for the three refrigerant properties of viscosity, specific volume, and refrigeration effect with the final chart entry constant as calculated.

Should anyone not care for the above assumptions or had a modified cycle involving multi-staging, intercooling, or temperatures outside the range covered, or any other different cycle conditions for whatever reason, he can still enter the chart by calculating his own value of $2.58 \times 10^5 \mu^{.2} v^2 \div (RE)^{2.8}$ and then use the vertical numerical scale for entry along the left hand side and ignore the refrigerant operating temperature conditions part of the chart.

It may be noted above that the viscosity parameter has a conspicuously weak effort since it varies from about 0.01 to 0.02 centipoises for most refrigerant vapors over a wide range of operating conditions, and then after taking the 0.2 power, the result comes close to 0.40 and 0.44 universally for both suction and discharge line vapors respectively for most of the usual refrigerants.

Specific volume squared is, on the other hand, a highly important parameter to obtain as closely as possible because of its wide range of variation with different operating conditions, which when squared increases the magnitude of effect still more. However, its value can easily be obtained with sufficient accuracy by reference to a standard Mollier (P-H) chart for the re-refrigerant used once the operating conditions have been established.

Refrigeration effect is also important since it is raised to the 2.8 power, but again the Mollier chart serves adequately and indicates a possible range of 40 to 60 for Refrigerant 12 (or 60 to 80 for Refrigerant 22 and about 420 to 520 for ammonia) over the range of operating conditions assumed above.

To illustrate further with a numerical example, assume an Refrigerant 12 system with R.E.=50 Btu/lb and specific volume = 4 cu ft/lb, and viscosity = .015 centipoises; then the chart entry constant would be $2.58 \times 10^5 \times (.015)^{.2} \times 4^2 \div 50^{2.8} = 31.1$. This would be true for smooth copper tubing as used typically with Refrigerant 12 and 22 systems, but if steel tubing were used, then the friction factor can be approximated at about 25% to 60% higher, according to some authorities (refer. 13 and 17) and assuming the 25% increase here, would result in a formula constant of about 3.22×10^5 instead of 2.58×10^5 and thus the chart entry constant would increase to 38.9 instead of 31.1. Ammonia systems, of course, use ferrous pipe only and there is no need for copper pipe information on the ammonia pipe selection chart.

For either the Refrigerant 12 or 22 charts where both copper and steel piping lines appear, the 25% difference in friction factor is allowed for by basing the chart entry constant on the friction factor for copper and then locating the price line for steel 25% lower than indicated by the pipe cost analysis procedure in the appendix. For ammonia systems the friction factor is figured directly at 25% higher in determining the effect of the fluid properties on the chart entry values.

Thus, any of the three charts can be used for any refrigerant other than Refrigerants 12 and 22, or ammonia and also for any modification of the given refrigerant cycle provided one allows for the proper pipe material friction factor and then calculates his own chart entry constant based on the three refrigerant properties corresponding to the refrigerant selected at the system cycle conditions as decided upon.

Each engineer would probably develop his own opinion as to suitable estimating values to use for E, L_e/L, $1/\eta_o$, and U_f. For instance, 2c per kw hr is close to the national average for electricity but this varies frequently from 1/4c to 5c per kw hr or more in different localities. The important consideration here would be to estimate the average price that the user of the system will be paying for electricity to operate his system under average installed conditions and taking into account the price schedule established by the local utility for such electrical usage and demand. Along this same line, when considering the price of pipe in order to locate the pipe cost line (as described in the appendix), it is important to estimate the actual price that the purchaser of the system is to pay for the piping as installed including the effect of possible price discounts for quantity orders, cost of fittings, and installing labor costs.

The overall efficiency, η_o, is the isentropic compression and motor efficiency combined and usually varies from about 65% isentropic compression efficiency $\times$ 70% motor efficiency, which equals 45% overall efficiency for small systems of less than one ton capacity, through approximately 80% $\times$ 85% = 68% efficiency for medium sized systems of 1 to 20 tons capacity, up to possibly as high as 90% $\times$ 95% = 85% efficiency for large systems of 100 or more tons capacity. This results in the reciprocal of efficiency values, $1/\eta_o$, ranging from about 2.25 to 1.2.

The usage factor, U_f, will vary from possibly 900 hr/yr (or about 10%) for air conditioning systems in northern U.S. climates through about one-third running time (or 33%) for average food refrigeration systems to possibly 75% or 80% for some special purpose, almost continuous running industrial air conditioning or refrigeration systems.

EFFECT OF PIPE LINE LENGTH ON SIZE SELECTION

The ratio of equivalent length of the pipe line system to the actual length (L_e/L) might fall in the range of 1.25 for a medium length or even a long piping system with

an average number of fittings consisting of say six elbows and one shut-off valve. Or L_e/L could be as great as 2 in a short piping system with a relatively large number of fittings, especially for closely coupled system of large pipe diameter. Thus, when the adjustment is made on the chart for some estimate of L_e/L, one is actually allowing for the increased energy cost due to the extra friction of fittings and valves. However, no allowance has yet been made in the analysis for the specific volume increase due to the total line pressure drop which results in an added increment of cost for the relative increase of compressor displacement requirement as pipe lines become longer.

A possible way to check the magnitude of this volume increase effect might be to estimate the added cost of a greater displacement compressor in terms of larger dimensions to just compensate for the volume increase due to pressure drop. Or another way would be to increase the compressor rpm sufficiently to handle the volume increase with no loss in system capacity and then shorten the compressor life a proportional amount. In either case, this would have the effect of a greater annual investment cost for the compressor compared to the cost of the pipe lines and could be allowed for by adding an increment of cost to the friction side (or the electrical cost term) of the equation.

For instance, over a wide range of evaporator conditions it can be shown (see appendix) that the amount of pressure drop per 100 equivalent feet of pipe length will increase the specific volume of vapor flowing in either a suction or discharge line by usually less than 10%. Then, whether a larger displacement compressor (dimensionally) be chosen to compensate for suction line Δ P, or if capacity be retained by say a 5% increase in rpm and hence a shorter operating life, either way results in about a 5% increase in compressor investment cost.

An investigation of a few price lists for compressors (only) indicates that a typical cost might be of the order of \$50 to \$100 per ton capacity; then on a 15 year life basis this would convert to an annual investment cost of about \$5 to \$10 per ton, and 10% of \$10 say would be \$1 additional compressor investment cost per year per ton for each 100 equivalent feet of pipe. Meanwhile, the electrical operating cost at a typical rate of 2c per kw hr and 1 kw hr per ton-hr as estimated for air conditioning operating conditions, and one-third running time annually would amount to an estimated annual operating cost of 2c $\times$ 3000 hr $\times$ 1 = \$60 per year per ton capacity. \$1 (or even \$1.20 say) would, at the most, be equivalent to not over 2% (and as low as 0.2%) in increased electrical operating cost for each 100 equivalent feet increase in pipe length. And the pipe selection charts clearly indicate this is a negligible quantity. This is, indeed, one of the major conclusions of this whole analysis; i.e., that the occasional practice of oversizing unusually long piping systems by one or two pipe sizes is out of proportion for optimum economy conditions.

TABLE II — COMPARISON OF PIPE SIZES BY ΔP ALLOWANCE VERSUS OPTIMUM ECONOMY

TON CAPACITY 100 F COND. (SUCTION LINES)

REF-RIG.	PIPE SIZE TYPE L-OD	—40 F EVAPORATOR ASRE $\Delta P = .49$	Opt. Econ. 2c/kwhr	ΔP	0 F EVAPORATOR ASRE $\Delta P = 1.01$	Opt. Econ. 2c/kwhr	ΔP	+40 F EVAPORATOR ASRE $\Delta P = 1.8$	Opt. Econ. 2c/kwhr	ΔP	Disch. Lines 100 F Cond. 0 F Evap. ASRE $\Delta P = 3.66$	Opt. Econ. 2c/kwhr	ΔP
	1/2	—	0.12	1.78	—	0.24	2.31	0.31	0.43	2.95	0.54	0.8	4.05
12	1 1/8	0.51	0.78	.81	1.39	1.75	1.29	3.10	3.10	1.71	5.30	5.4	2.2
	3 1/8	2.94	3.60	.41	22.70	18.00	0.54	49.5	34.0	0.76	85.00	55.0	0.92
	6 1/8	50.80	42.00	.24	137.0	91.0	0.34	299.0	160.0	0.42	494.0	265.0	0.54
	Steel Sch. 40												
	1	0.46	.64	.51	1.22	1.35	0.72	2.68	2.5	0.98	4.35	4.2	1.24
12	3	8.04	7.60	.25	20.6	15.0	0.33	43.8	28.0	0.44	70.8	47.0	.56
	6	48.2	35.0	.15	124.0	73.0	0.21	268.0	135.0	0.28	421	220	.35
	12	286.0	165.0	.10	729.0	335.0	0.13	1520.0	600.0	0.17	2460	1000	.21
	Steel Sch. 40	$\Delta P = .32$			$\Delta P = .78$			$\Delta P = 1.48$			$\Delta P = 3.3$		
	1	—	1.8		3.46	3.9	.90	7.5	7.3	1.4	15.9	12	1.98
NH_3	3	26.9	20.0	.19	73.9	42.0	.38	162	77.0	.39	338	120	.51
	6	160.0	92.0	.12	445	200	.24	958	360.0	.25	1986	570	.37
	12	975.0	430.0	.074	2640	920	.14	5680	1700.0	.17	—	2700	

NOTES: ΔP values are psi per 100 feet equivalent length of pipe.
For optimum economy basis: $U_f = 30\%$, $L_e/L = 1.25$
$\pi_e = 50\%$ for small systems and
$\pi_e = 88\%$ for large systems

It should be pointed out here that compressor speed changes are frequently impossible to make due to the increasing popularity of direct drive motor-compressor units. Furthermore, small incremental adjustments of compressor displacement by dimensional changes (as low as 5% steps) are almost never encountered within the line of compressors available from a single manufacturer. Hence, even though the methods suggested above for adjustment of compressor displacement are seldom feasible, yet it is one way to approximate the relative worth of refrigerant volume increase due to pressure losses in pipelines.

Reasons other than optimum economy often influence the selection of pipe size. For instance, sizing to assure adequate oil return, especially at part load conditions needs to be checked (see chap. 53, Reference 2). Also, system capacity balance can frequently be improved by slightly oversizing lines to result in some effective gain in compressor capacity; or conversely, an undersized line will somewhat limit the capacity of an oversize compressor when the incremental size, as mentioned above, cannot be obtained to exactly match the refrigeration load. Then sometimes piping systems are purposely undersized to result in a more favorable contract price where competitive bidding is involved.

For this last situation a definite precaution should be observed by reference to Fig. 1 which is constructed approximately to scale for a hypothetical system. Note the optimum diameter would be about 4 in.; the next size smaller (3½ in.) would not greatly increase the cost to the buying customer by more than 5 to 10%; but the next size smaller (3 in.) would greatly increase the cost to the customer by about 50% more in overall owning and operating expense as compared with the optimum 4 in. size. Oversizing lines, on the other hand, is not nearly as serious in ultimate overall cost to the customer, although the investment cost as shown by the equipment contract might appear to be quite unfavorable.

RELATIVE EFFECT OF PARAMETERS ON PIPE SIZE

Finally, consider the effect of the remaining parameters on pipe size, some of which do have a considerable effect and are seldom allowed for in present pipe tables. Inspection of the pipe charts here will indicate that it takes about a 3 times multiplication of any vertically located parameter to require a change of one pipe size. Thus, use factor, or electrical cost, or efficiency, or equivalent length ratio, or installed pipe cost would each have to change by a factor of 3 individually, or by a collective factor of 3 due to any total combined effect, to cause one pipe size change. Similarly it takes about 15 to 20 F change in evaporator temperature to call for one pipe size change of suction line, but the same is not true for discharge lines where evaporator temperature variation has a minor effect.

Changes in condensing temperature have a minor effect in

both lines and it is of some interest to note that what little effect there is results in opposite trends for suction lines compared to discharge lines. Finally, it takes about a 1½ to 2 times change in capacity or tonnage to require one pipe size change.

COMPARISON WITH OTHER PIPE SIZING METHODS

A comparison now of the pipe sizing method presented here with existing pipe selection tables is of interest. Table II shows such a comparison based on excerpts from the pipe sizing tables in the latest editions of the ASRE Data Books (Design Volume 10, 1957-58, and Application Volume 1, 1959). These tables are from ARI data converted to an equivalent Δ t from the original pressure drop basis. It may be noted for the mid-range of pipe sizes (1 to 3 in.) that either method would indicate about equal capacity recommendations. But in the larger sizes of systems the optimum cost analysis would call for progressively larger diameter piping than is provided by the ARI schedule.

Comparison in Table II is based on the following assumptions of reasonably typical or average values: Electricity at 2c per kw hr, use factor of 30%, L_e/L ratio of 1.25, and overall efficiency appropriately graduated with size of system from 50 to 85%.

While different pressure loss allowances are often presented (references 1, 2, 11, 12, 14, 15, 16, 18, 20), the exact value to use in a given design situation is left up to the engineer, and little or no information is given accompanying such tables as to the relative effect of the important economic parameters that should enter into the design selection. Such parameters as electrical cost, installed pipe cost, use factor, and system efficiency may all have a combined effect of one or two pipe size difference, which would require a sliding scale of pressure drop values to allow for these several variable factors.

Fig. 6 shows how the pressure drop allowance and also the Δ t equivalent should vary with pipe diameter for optimum economy (assuming the conditions listed above). It also shows the pressure drop required to maintain a velocity for adequate oil return in vertical upfeed risers (according to Holladay's analysis Ref. 15) and indicates a considerable safety factor in this respect of oil return for all sizes of Refrigerant 12 lines and at all temperatures except for evaporator temperatures below − 40 F where there is little or no margin of safety.

PIPE SIZING FOR LIQUID LINES

Pipe sizing for liquid lines is not a question of optimum economy at all, but rather one of available pressure drop resulting from subcooling the liquid in the condenser or in any heat exchange device between the condenser and expansion valve where subcooling takes place (see References 2, 11, 12, 14, 17). The basis for design of liquid lines is that as long as the total pressure drop (due to friction of the fluid

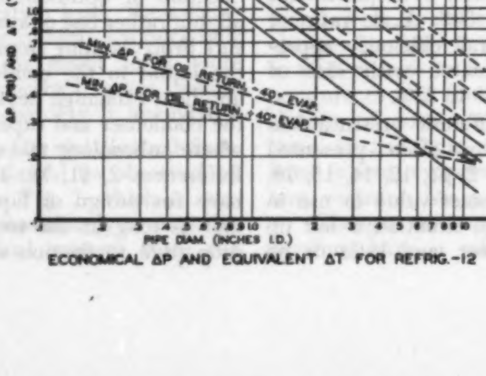
flowing in straight pipe plus fittings as well as any head loss due to rise of liquid in any vertical runs) does not exceed the pressure drop equivalent for the degrees of subcooling of the liquid as it leaves the condenser and before it gets to the expansion valve, then the design will be satisfactory; i.e., formation of flash gas before the expansion valve will be avoided, and hence the capacity of valve will not be greatly reduced by pressure drop. Consequently, the allowable pressure loss method is the accepted way to size liquid lines. And since most condensers will subcool the liquid 5 to 10 F, a Δ t allowance of say 2 F per 100 equivalent feet for Refrigerant 12 liquid line pressure drop should be adequate and allow some safety factor usually. This corresponds to 3.6 psi Δ P allowance for Refrigerant 12 as listed in the liquid line sizing tables in the recent ASRE Data Books. (1 F Δ t allowance = 3.3 Δ P recommended for ammonia.) (References 1, 2).

SUMMARY AND CONCLUSIONS

1. The analysis presented here includes the major factors which should influence pipe size selection. Some engineering skill and judgment is still needed to estimate suitable values for the several parameters in each design situation. However, the method here is quite flexible so the engineer may easily change conditions or compare alternate assumptions and note the relative effect or magnitude of any parameter change in question.

2. The customary pressure drop rules ranging in value from ½ to 3 psi per 100 ft for pipe sizing as

Fig. 6



ECONOMICAL ΔP AND EQUIVALENT ΔT FOR REFRIG.-12

well as the Δ t rule of 1 to 2 F per 100 ft appear to result in economic pipe diameters for the mid-range of 1 to 3 in. sizes where typical present-day prices are assumed. But for the larger diameters the optimum cost method here would call for about one pipe size larger than would the recent ASRE tables based on one value of pressure drop (of say 2 psi or 2 F Δ t equivalent for Refrigerant 12) over the entire range of pipe sizes. Further, such pressure drop (or equivalent Δ t) rules, which have been presented in most reference tables easily available to the designer, have seldom stated the economic conditions which were assumed in establishing the pressure drop values or what adjustment to make in the pressure drop allowance for possible changes in electrical, material, or labor costs; nor have such factors as system usage (hr operating time per yr) or overall compressor-motor efficiency been allowed for.

3. The frequently quoted rule of one or two pipe size increase for unusually long lines would not appear necessary from an optimum economy standpoint. The actual length of piping system should have little or no effect on the diameter selection. However, equivalent length to actual length ratio can have a minor effect, but since a multiplying factor of 3 (approximately) is required to call for one pipe size change (from the results of this analysis), it is difficult to conceive of a piping system with enough fittings in proportion to length to warrant a full pipe size increase.

APPENDIX

CONSTRUCTION OF FIG 3, 4, 5, AND 7

To locate the pipe cost line on the pipe selection charts, the following method was used for copper tubing.

1. First, a graph was made of cost per ft of straight tubing versus pipe diameter on an actual I.D. basis. A 1955 price list was used and quantity purchase of 10,000 ft lot size was assumed. This located curve A in Fig. 7.

2. In order to estimate the 1961 price, the ENR index for refrigeration equipment was found to be about 211 for 1955 (Reference 22) and rising at about 17 points per year. Thus by 1961 the ENR index could be estimated at approximately 313. So curve B was located at about 313/211 = 1.5 or 50% higher than curve A.

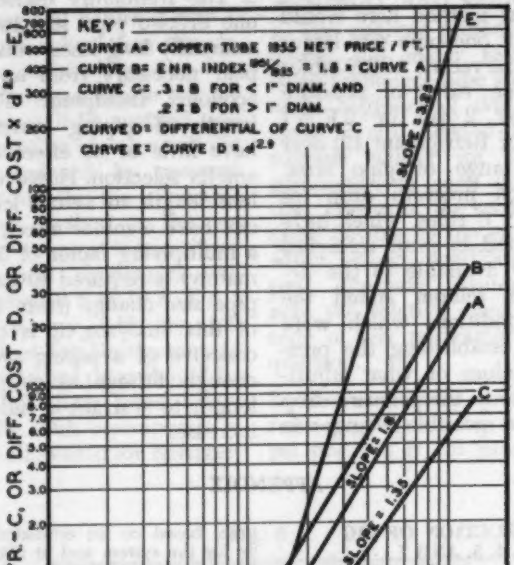
3. Installation costs have been estimated at from 2 to 5 times the cost of straight pipe alone. Here a value of 3 times was assumed for sizes up to 1 in., and a value of 2 times at 6 in. diam. At the same time, based on an estimated life of 15 yr for the system and at 6% interest on the investment, the annual investment cost would be approximately 1/10 of the original installed cost. This located curve C at 3/10 of curve B for sizes less than 1 in. and 2/10 of B at 6 in. diam. Note that the slope of the curve C on log log plot is now 1.25 at less than 1 in. and 1.35 at over 1 in. diam.

4. Now to differentiate curves which are straight lines on log log plots, one reduces the original slope by one and multiplies the ordinate value at the intercept of unity on the abscissa scale by the slope, n, of the original curve. In this case, the differential of curve C to the left of 1 in. has a slope of $1.25 - 1 = 0.25$ and crosses unity at $.20 \times 1.25 = .25$; and to the right of 1 in. the slope of curve D has a slope of 0.35 and intercepts unity at $.20 \times 1.35 = .27$. Thus, curve D is the differential of curve C and could be transferred directly to the pipe selection chart as the solution line.

5. However, a few chart modifications

were next made simply to improve the appearance, make the scales easier to read, and make the guide lines easier to follow. Since the friction term in the differentiated equation varies in magnitude inversely as d to the 5.8 power, one half

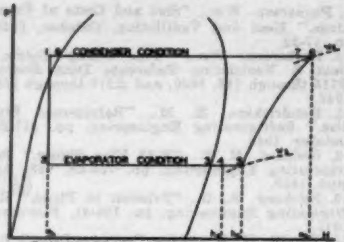
Fig. 7



of this power, or d to the 2.9 power, was taken as a multiplier of both terms. This makes the friction term vary inversely to the 2.9 power, and the differentiated pipe cost curve varies directly to the $0.25 + 2.9 = 3.15$ power and $0.35 + 2.9 = 3.25$ power as shown for the slope value of curve E at values less than and greater than 1 in. diam, respectively. Finally, if the diameter scale (abscissa) for the pipe selection chart is chosen as 2.9 times the ordinate scales, then the guide lines will be at a slope of exactly 45° and the pipe cost solution line will be at a slope of slightly over 45° corresponding to slopes $3.15 \div 2.9 = 1.08$ and $3.25 \div 2.9 = 1.14$ respectively to the left and right of 1 in. diam. Without the above modifications the guide lines would slope at a value of nearly 6 to 1 or after multiplying by $d^{2.9}$ the slope would be 2.9 to 1 which is still too steep for easy reading and results in very short diameter scale along the abscissa.

EFFECT OF PRESSURE DROP PER 100 FEET ON SPECIFIC VOLUME INCREASE

To check the effect of specific volume increase due to pressure drop per 100 ft of pipe line, refer to Fig. 6 again for the economical pressure drop recommendations. Note that for a 40 F evaporator and small pipe sizes the ΔP per 100 ft of pipe is of the order of 3 psi for Refrigerant-12, which from saturated refrigerant vapor table results in a ΔV of about .04 at a specific volume of .75 or about 5.3% volume change per 100 ft. At larger pipe sizes the recommended ΔP would be about 1/10 of the above value or ½ volume increase. Then at −40 F evaporator, the ΔP per 100 is about 2 psi which corresponds to a volume change of about .35 at a specific volume of about 3.5 cu ft/lb. This, then, is about a 10% volume increase for each 100 ft of pipe line and



P-H CHART OF REFRIGERATION CYCLE

Fig. 8

1/10 of this value or 1% for larger pipe sizes.

CONSTRUCTION OF FIG 1

Fig. 1 was constructed in the following manner: First, curve B was located to represent the installed cost of piping according to a given price schedule; it is actually the same as curve C in Fig. 7. Then, it was assumed there could be a tonnage, use factor electric cost, L_e/L ratio, and overall efficiency combination such that the pipe friction cost line would go through the arbitrarily chosen point located a $2 cost and 3 in. diam. Other values of cost versus diam. were then calculated from the total cost equation to properly locate curve A with respect to this first arbitrary point. Finally, A and B were added to determine curve C. There would actually be an infinite number of B curves covering all possible values and combinations of T, U_f, E, L_e/L and η_o. It is of interest to note the flatness of curve C in the optimum region over a range of about one pipe size on either side of the optimum; then, the rather gradual rise in curve C for larger pipe sizes, but the quite rapid rise in C for two or more pipe sizes smaller than optimum.

BIBLIOGRAPHY

1. ASRE Design Data Book, 10 Edition, 1957-58, Chapter 9.
2. ASRE Applications Data Book, Volume 1, 1959, Chapters 52, 58.
3. Christian, T. R., "Estimating Pipe Costs," Heat, Pipe & Air Conditioning, June, 1954, pp. 102-6.
4. Zimmerman & Lavine, "Chemical Engineering Costs," Chapters 1 and 20, (on piping costs), Industrial Research Service, Dover, New Hampshire.
5. Chemical Engineering, November, 1947 and January, 1950, "N Factor for Pipe Costs."
6. Larian, M. G., Fundamentals of Chemical Engineering Operations, (on optimum pipe size calculation), pp. 29-36, Prentice Hall, 1958.
7. Doremus, R. C., "Refrigeration Piping," Heat Piping & Air Conditioning, January, February, 1958.
8. Gygox & Wilson, "Pressure Drop in Refrigeration Liquid Lines," Refrigeratting Engineering, February, 1940.

9. Parkerson, Wm., "Size and Costs of Freon Lines," Heat and Ventilating, October, 1940, pp. 17-22.
10. Parkerson, Wm., "Pipe Sizing Charts," Heat & Ventilating Reference Data Sheets, #115 through 118, 1940, and #217 through 222, 1942.
11. Hendrickson, H. M., "Refrigerant Pipe Size," Refrigerating Engineering, pp. 317-25, October, 1946.
12. Goddard, M. B., "F-12 Pipe Sizing," Refrigerating Engineering, pp. 753-58, 763, August, 1950.
13. Stickney, A. B., "Friction in Pipes," Refrigerating Engineering, pp. 129-31, February, 1947.
14. Ferrill, H. E., "Line Pressure Drop," Refrigerating Engineering, pp. 425-67, November, 1946, pp. 486-7, November, 1947, also "Pipe Sizing Heating Pipe & Air Conditioning," March, May, October, 1947.
15. Holladay, W. L., "Pressure Drop in Freon Systems," Refrigerating Engineering, pp. 55-61, 96, September, 1954.
16. Districk, S. C., "Refrigeration, Piping for Air Conditioning," Heating and Ventilating, Vol. 50, pp. 95, March, 1953.
17. Carrier Engineering Design Data, Sections 25, A, B, and X on Piping, 1946-53.
18. Jordan and Priester, Refrigeration and Air Conditioning, Chapter 12 on Piping, Prentice Hall, New York, 1956.
19. Stoecker, W. F., Refrigeration and Air Conditioning, Chapter 23 on Piping, McGraw-Hill, New York, 1958.
20. ACRMA Equipment Standards, #510 and #520, 1946.
21. Sutton, G. E., Analysis of Refrigeration Systems, Chapter 5 on Piping; unpublished Ph.D. Thesis, Michigan State University, 1957.
22. Chemical Engineering, ENR Index for Refrigeration Equipment, February, 1959.

DISCUSSION

RICHARD LINE, St. Louis, Mo.: The evaluation did not consider systems where the compressor runs about 100% of the time. In these instances, if the pipe size were dropped, it might be necessary to put on the next size motor and compressor.

AUTHOR RENWICK: If actual operating time was 100%, the use factor would also be 100% and no adjustment would be required for the use factor in the pipe selection charts. The cost of volumetric displacement of larger compressors to compensate for pipe friction was analyzed in the paper and is considered negligible in effect; discussion of this point was omitted in the oral presentation for lack of time.

JOHN DUNHAM, Pittsburgh, Pa.: Was the liquid line sizing not evaluated because pressure is limited from the flashing?

AUTHOR RENWICK: Yes. The liquid line sizing is an entirely different problem for several reasons. According to Chapter 32 or 53 of the last DATA BOOK, liquid line sizing should still be done on pressure-drop basis. Some will maintain that, as far as the refrigeration process is concerned, it does not matter whether pressure-drop occurs in the expansion valve or the liquid line. It is realized, however, that if some pressure is lost before reaching the expansion valve, less liquid will go through, which limits the capacity of the valve.

The important point is that flash gas be avoided by the time the expansion valve is reached. As long as the condenser can create enough subcooling, for example 9, 7 or 10 F, which in pressure-drop is equivalent to 10 psi for Refrigerant 12 and as long as the pressure-drop is not used up through any friction in the fittings, elbows or valves, there is no problem with the flash gas.

The solution generally is to take the expected degree of subcooling the condenser can maintain and convert that to equivalent pressure-drop; roughly divide by two for a safety factor; and design the liquid line on the remaining pressure-drop, after allowing for an estimate of the expected pressure-drop for pipe elbows and other fittings and also for any anticipated vertical lift distance in the liquid line.

1745

No. 1745

Graphical Analysis of a Cross-Flow Cooling Tower

HIDEO UCHIDA
Member ASHRAE

Analyses of cross-flow cooling towers have been provided previously by several authors. Snyder's theoretical analysis, assuming a linear relationship between water temperature and the corresponding enthalpy of saturated air, resembles that of cross-flow heat exchangers, except that enthalpy potential is used for cooling towers, whereas temperature is used for heat exchangers. It was Snyder's purpose to obtain experimental results. Zivi and Brand use the same differential equation presented in this paper giving the water temperature distribution in the fill of a cross-flow cooling tower. This procedure is suited for automatic computation under the trial and error method.

GRAPHICAL ANALYSIS

Fig. 1 is a schematic vertical section through the direction of the air flow. Water entering the top of the tower flows uniformly and at the same temperature down through the fill. Direction of the air flow is defined as positive x-direction while the positive y-direction is the direction of water flow. Total length of the air flow and water flow are X and Y, respectively. Air conditions, temperature and enthalpy are the same along the entire length Y. Through any z-direction which is perpendicular to both x and y, the water temperature and the air enthalpy are assumed to be the same. Where the width of the tower is Z, the analysis of cross-flow cooling tower can be treated as a part of a two-dimensional problem.

Incremental element of volume is shown in Fig. 1. Water temperatures entering and leaving the element of volume are t_{wxy} and $t_{wx,y+\Delta y}$, respectively. Enthalpies of saturated air, the thin air films

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over the surface of water entering and leaving the element of volume are i'_{wxy} and $i'_{wx,y+\Delta y}$, respectively. Air enthalpies entering and leaving the element of volume are i_{xy} and $i_{x+\Delta x,y}$.

Heat quantity transferred between air and water in the incremental element of volume, the symbol ΔQ given the following relationships.

$$\Delta Q = \frac{L}{X}\,\Delta\chi\,(t_{wxy} - t_{wx,y+\Delta y}) = -\frac{L}{X}\,\Delta\chi\,\Delta t_{wxy} \qquad (1)$$

$$= \frac{G}{Y}\,\Delta y\,(i_{x+\Delta x,y} - i_{xy}) = \frac{G}{Y}\,\Delta y\,\Delta i_{xy} \qquad (2)$$

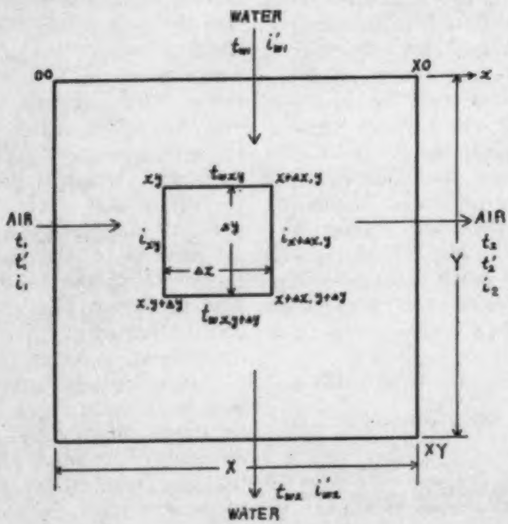
$$= KaZ\,(i'_w - i)_{mean}\,\Delta\chi\,\Delta y = KaZ\,\Delta\chi\,\Delta y\,\Delta i_{xym} \qquad (3)$$

$$\doteqdot \tfrac{1}{2}\,KaZ\,\Delta\chi\,\Delta y\,[(i'_{wxy} + i'_{wx,y+\Delta y}) - (i_{xy} + i_{x+\Delta x,y})] \qquad (4)$$

Assuming constant overall heat transfer coefficient Ka in the tower, the transfer unit U of a cross-flow cooling tower can be represented thus:

$$U = \frac{KaV}{G} = \frac{1}{Y}\sum \frac{\Delta i_{xy}}{\Delta i_{xym}}\,\Delta y \qquad (5)$$

Fig. 1 Vertical section through direction of air flow



$$\frac{U}{N} = \frac{KaV}{L} = \frac{1}{X} \sum \frac{-\Delta t_{wxy}}{\Delta i_{xym}} \Delta\chi \qquad (6)$$

Using integral forms, those relations are

$$U = \frac{1}{Y} \iint \frac{\dfrac{\partial i}{\partial \chi}}{i'_w - i} \, d\chi \, dy \qquad (7)$$

$$\frac{U}{N} = \frac{1}{X} \iint \frac{-\dfrac{\partial t_w}{\partial y}}{i'_w - i} \, d\chi \, dy \qquad (8)$$

Fig. 2 represents a part of the t-i diagram of humid air. Intersection of the t_{wxy}-line and the i_{xy}-line is point α (x,y). Point γ, the intersection between the $t_{wxy+\Delta y}$-line and the i_{xy}-line. Point δ is the intersection between the t_{wxy}-line and the $i_{x+\Delta x,y}$-line, and point β the intersection of the $t_{wx,y+\Delta y}$-line and the saturation curve. The direction of the line γ-δ can be represented, dividing eq. (1) with eq. (2).

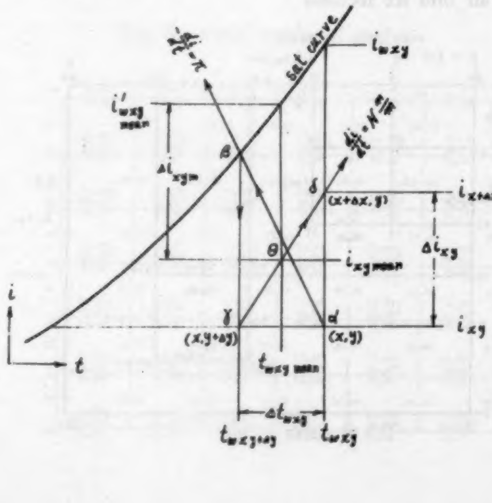
$$\frac{di}{dt} = \frac{\Delta i_{xy}}{-\Delta t_{wxy}} = \frac{L}{G} \frac{\Delta\chi}{X} \frac{Y}{\Delta y} \qquad (9)$$

Dividing length X into m equal sections of length $\Delta x = X/m$, and the length Y into n equal sections of length $\Delta y = Y/n$, eq. (9) is expressed as

$$\frac{di}{dt} = N \frac{n}{m} \qquad (10)$$

Therefore, when the tempera-

Fig. 2 Part of the t-i diagram of humid air



ture drop of water through the incremental element of volume is known $\Delta t_{wxy} = [t_{wxy+\Delta y} - t_{wxy}]$ Δt_{wxy} will be found negative, and with the given water-air ration N, point δ can be obtained on the line γ-δ with a direction. $\frac{di}{dt} = N\frac{n}{m}$.

Then the water temperature $t_{wx,y+\Delta y}$ and the air enthalpy $i_{x+\Delta x,y}$ can be found, and the element of volume remains.

Average conditions of air and water in the incremental element of volume are point θ in Fig. 2, the middle point of the line γ-δ. Reading the values of the point θ, it is possible to get the average enthalpy of air i_{xym}, the average temperature of water t_{wxym}, and the average enthalpy of saturated air over the water i'_{wxym}. Enthalpy difference between the air and saturated air over the water, Δi_{xym}, the enthalpy potential difference in the incremental element of volume.

We refer π to the $\frac{di'}{dt}$ of the direction of the line α-β as follows:

$$\pi = \frac{\beta\gamma}{\gamma\alpha} = \frac{i'_{wx,y+\Delta y} - i_{xy}}{t_{wxy} - t_{wx,y+\Delta y}} \qquad (11)$$

Putting $\beta\gamma \doteq \Delta i_{xym}$, in differential triangle αγδ produces the following approximate relation to the eq. (6)

$$-\frac{di'}{dt} = \pi \doteq \frac{mN}{U} = \frac{mL}{KaV} \qquad (12)$$

As seen in eq. (12), π is an approximate definite value in the tower. Therefore, if we get point β, which is the intersection between the saturation curve of t-i

Fig. 3 Division of flow length of water and air into six sections

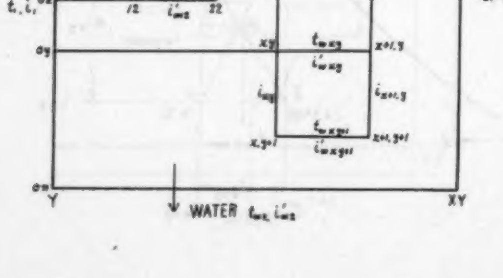


diagram and the line of the direction $-\pi$ drawn from the point α, we can get the point γ from the point β and the point δ from the point γ. Then, it is possible to have the water temperature and the air enthalpy entering the next incremental element of volume. When this procedure is continued, alternately, as shown in Fig. 4, the distribution of water temperatures and air enthalpies in the tower can be determined.

Substituting those values obtained through the procedure into eq. (5) and (6), approximate values of U, U/N and Ka, respectively, can be found, and using the larger number of m and n, the more exact solution of U/N and Ka can be determined.

Example of graphical solution – Inlet air enthalpy $i_1 = 38.54$ (corresponding to the inlet wet-bulb temperature of air $t_1' = 82.04$ F), inlet water temperature $t_{w1} = 98.6$ F and water-air ratio $N = 1.2$ are the given conditions. The problem is to find the values of U/N or Ka in order to make the outlet water temperature $t_{w2} = 87.0$ F under those given conditions.

Putting $m = 6$ and $n = 6$, assuming $\pi = 6/1.83 = 3.28$. The selected value 1.83 will differ

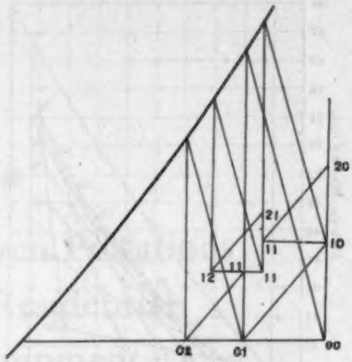
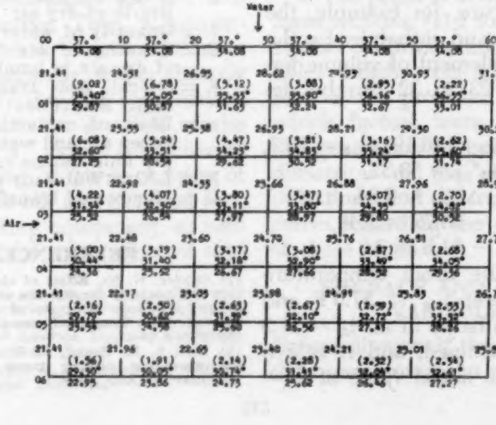


Fig. 4 Part of the graphical analysis of the tower

Fig. 5 Final results of analysis



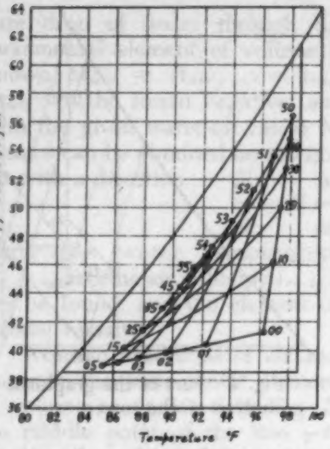


Fig. 6 Average conditions of water and air in tower

slightly from the real value of U/N. Fig. 3 divides the flow length of water and air into 6 sections. Fig. 4 shows a part of the graphic analysis of the tower; line $\alpha\beta$ with di/dt $= -3.28$, and the line $\gamma\delta$ with di/dt $= 1.2$ can be drawn by the same procedure presented in Fig. 2. The final result is represented in Fig. 5. From the figure, for example, the temperatures and enthalpies for the incremental element of volume figured by 22-23-33-32 can be determined.

$i_{23} = 43.79$, $i_{33} = 46.19$, $t_{w22} = 93.5$,
$t_{w23} = 91.5$, $i_w{'}_{22} = 53.32$,
$i'_{w23} = 50.35$, $\Delta i_{22m} = 6.84$,
$\Delta i_{22} = 46.19 - 43.79 = 2.40$,
$\Delta t_{w22} = 93.5 - 91.5 = 2.0$

From this the mean temperature of outlet water $t_{w2} = 87.0$ F and the outlet enthalpy of air $i_2 = 51.3$ can be determined, and it is evident that the initial value of $\pi = 3.28$ was good, for it resulted in $t_{w2} = 87.0$ F.

Substituting those values shown in Fig. 5 into eq. (6), the U/N can be calculated as 1.765. The real value of U/N = 1.765 is the difference of 3.7% from the value as 1.83 given in π. If we select any value for V/L, the value of Ka, which represents the performance of the tower, can be obtained, substituting U/N = 1.765 into eq. (6).

Fig. 6 shows the average conditions of water and air in the tower.

NOMENCLATURE

Y = Length of air flow perpendicular to water flow
X = Length of water flow perpendicular to air flow
Z = Width of tower perpendicular to X and Y
V = Total volume of tower
t = Dry bulb temperature of air, F
t' = Wet bulb temperature of air, F
i = Enthalpy of air, Btu per pound of dry air
t_w = Temperature of water, F
i'_w = Enthalpy of saturated air temperature of which is t_w, Btu/lb of dry air
L = Quantity of water flow, lb/hr
G = Quantity of air flow, pounds of dry air in humid air per hr
K = Overall heat transfer coefficient, Btu/ft²hΔi
a = Ratio of contacting area between air and water, unit area per unit volume of tower
N = L/G = Water air ratio
U = Number of transfer unit

REFERENCES

(1) Snyder, N. M., Effect of air rate, temperature and packing density in a cross-flow cooling tower, American Institute of Chemical Engineering, Heat Transfer Symposium, vol 52, no. 18, 1955, p 61-67.
(2) Zivi, S. M. Brand, B. B., An analysis of the cross-flow cooling tower, Refrigerating Engineering, Aug. 1956.

1746

No. 1746

Combustion-Driven Pulsations in Oil-Fired Residential Heating Equipment

A. A. PUTNAM C. F. SPEICH

Pulsations in oil-fired equipment have long been a source of annoyance to the heating industry and its customers. Through the trial and error of past experience and limited research, a number of techniques have been devised which may suppress the occasional pulsations found in service units. However, the techniques are not uniformly effective. Thus, a measurable amount of industry effort is expended annually, either directly or indirectly, for the suppresssion of pulsations. As a result of the need for a more fundamental understanding of the mechanism of pulsations in oil- and gas-fired residential heating equipment, as well as the need for more effective suppression techniques, three sponsoring groups* initiated a research project at Battelle Memorial Institute. This paper, one of a series resulting from this program, covers the proposed mechanism of pulsation in oil-fired units. A subsequent paper will discuss suppression techniques for the same class of units.

The paper is divided into three main sections: (1) a summary of the studies of furnace and burner variables which were investigated as part of the program to determine which factors were important to the generation of pulsations, (2) a discussion of the mechanism of pulsation, and (3) concluding remarks regarding the consequences of these studies.

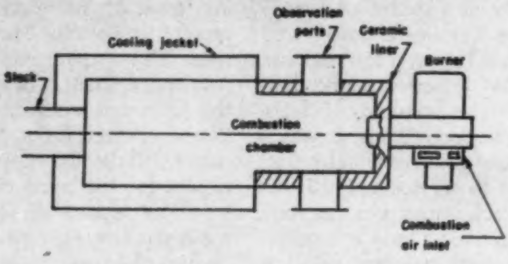
FURNACE AND BURNER VARIABLES

Before discussing the experimental data concerning the effect of vari-

* The sponsoring groups of this project were predecessor American Society of Heating and Air-Conditioning Engineers, Inc., the Oil-Heat Institute of America, Inc., and the American Gas Association.

A. A. Putnam and C. F. Speich are with the Battelle Memorial Institute.

Fig. 1 With this facility Battelle researchers studied oil furnace flame pulsations



ous furnace and burner variables on the pulsation amplitude, a short description of the procedure and equipment used for these studies is given.

Research Procedure and Equipment—In the beginning of this research program, a series of systematic studies on furnace and burner variables, using one experimental and several commercial furnaces, was made to determine how each affected the amplitude of pulsation. Different burners having solid and hollow air patterns were fired in the experimental furnace.

Fig. 1 presents both sectional and external views of the experimental furnace built in the early part of the program and used for most of the subsequent investiga-

tions. This axially symmetric furnace was designed to have a firing capacity of 0.85 to 1.00 gph. The internal simplicity of the furnace made it easier to calculate such factors as the natural acoustic frequencies of the furnace-burner system.

The Variables—Table I summarizes the effect of each of the furnace and burner variables studied in this and previous research programs. The Table is divided into two parts: (A) those variables which had an important effect on the amplitude, and (B) those which had little or no effect. Whenever possible, the effect of one variable was evaluated while all others were held constant.

It is seen from this table that the fuel spray, air, and recirculation patterns are the variables which are important to the generation of pulsations. Other variables such as draft, fan characteristics, and furnace configuration within normal limits have little effect on

TABLE I

SUMMARY OF THE EFFECTS OF VARIOUS FURNACE AND BURNER VARIABLES ON THE AMPLITUDE OF PULSATION

Variable	Effect
A. Variables having an important effect on the amplitude	
Fuel spray (angle and pattern)	— high- and low-amplitude levels were found (Fig. 2); air-fuel mixtures at which transitions occurred were found to be a function of fuel-spray angle
Air pattern	— affected the mixtures at which the transitions occurred
Recirculation pattern	— certain changes in the recirculation pattern were found to increase the amplitude
Acoustical characteristics of the blast tubes	— can either increase or decrease amplitude depending on the natural frequency of the furnace-burner system and the mode of oscillation
B. Variables having little or no effect on the amplitude	
Continuous ignition	— reduced the amplitude slightly in most cases
Burner fan characteristics	— can lend instability to the burner-furnace system
Draft	— affects the acoustical system by changing the natural frequency and the acoustic losses
Frequency	— transition points did not occur at a constant frequency
Furnace configuration	— can affect recirculation patterns, acoustic losses, or natural frequencies
Atmospheric conditions	— unknown

the generation of pulsations. The table indicates only the principal items which were studied in this and other programs. Investigations of other variables have been reported in the literature. However, since they have been found to have an insignificant effect on the pulsation amplitude when all other factors are held constant, these variables have been excluded from the table.

Fig. 2 summarizes the general results reported[1] for a burner with a solid air pattern. In the initial studies, the principal emphasis was placed on the effect of spray pattern on the amplitude of pulsation, because the spray pattern was both easily changed and had been found to have a significant effect.

It was found there were two ranges of pulsation, a high-amplitude range and a low-amplitude range. In each range, the amplitude gradually diminished as air-fuel ratios were increased. However, as the mixture was changed from rich to lean, the amplitude changed suddenly from low to high; then, after a further increase in air-fuel ratio, the amplitude dropped back to a low level. The mixture at which these rapid changes in amplitude occurred was found to be a function of the mean fuel-spray angle.*

* In the analysis of the fuel-spray pattern data, it was found convenient to replace the nominal designation of fuel sprays, namely, spray angle and pattern, with a hypothetical spray-pattern model. This model defined all sprays, regardless of angle or pattern, as a thin, hollow spray in which all of the fuel was concentrated at some effective angle. The model's effective angle of spray was defined, mathematically, as the mean angle of mass concentration of the original spray and was calculated in a manner similar to the determination of the center of gravity for a linear distribution of masses.

Fig. 2 How low- and high-amplitude ranges of pulsation related to a solid air pattern and typical amplitude observations for one nozzle

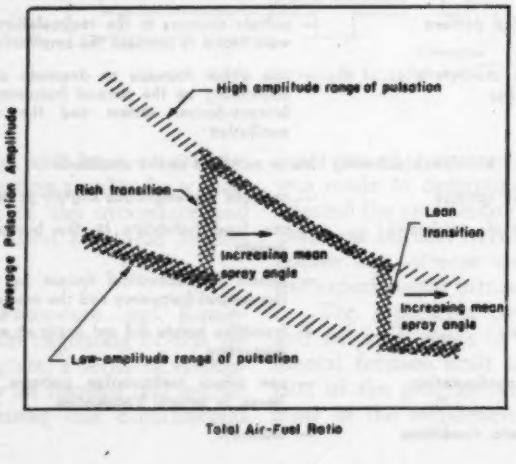


Fig. 2 represents a composite of the data for a number of different nozzle angles and types, and therefore covers a wide range of air-fuel ratios. With any specific nozzle and a reasonable operating range of air-fuel ratios, only part of the total pattern would be evident. Hence, for one nozzle, only the amplitude pattern around the rich transition point might be observed; for another, only the amplitude pattern around the lean transition point would be evident. This indicates the apparent contradiction encountered by many servicemen in attempting to eliminate undesirable pulsations by air adjustment alone; it is seen that it is possible to avoid pulsations in some circumstances by an increase in air-fuel ratio, and in other circumstances by a decrease in air-fuel ratio.

When the amplitude studies were extended to other burners having hollow air patterns, it was found that the results were similar, but not identical to those found with burners having solid air patterns. The observed effect on the rich-side transition points was found to be essentially the same, regardless of air pattern, for comparable changes in the mean fuel-spray angle. On the lean-side transition point, the transition point was found to move to leaner mixtures for increases in mean-fuel-spray angle when the air pattern was solid, but for hollow air patterns, the reversed trend was found to exist.

The effect of changes in the recirculation pattern in the combustion chamber was quite difficult to define in a manner that would lead to definitive results. However, two observations are worthy of mention. It was revealed that projecting the blast-tube exit into the combustion chamber increased the amplitude of pulsation. The results of a previous investigation[3] had shown that placing a disk on the axis of a combustion chamber, at the proper distance from the nozzle, also increased the amplitude of pulsation. It is concluded that the influence of both of these changes was felt primarily in the recirculation zone, and that the flow in this zone could affect the pulsation amplitude.

It was observed during this program that the pulsation amplitude was not the same among the various burners studied. Consideration of the available information indicated a correlation could be made between these changes in amplitude and the measured differences between the various natural frequencies of the entire furnace-burner unit and the single natural frequency of the burner considered separately. The difference in natural frequency, as well as any change which might occur in the specific mode of oscillation of the gases within the combustion chamber and blast tube, would affect the periodic flow pattern in the combustion region near the blast-tube exit, if the furnace-burner natural frequency remained essentially constant. Hence, the coupling between the acoustical system and the combustion system would be changed, thus changing

the amplitude of pulsation. This argument follows closely the principle outlined in a previous discussion of the mechanism of oscillation in gas-fired multiple-port burner-heating units.[3]

Research during the early part of the program brought to light the potential problem of room acoustics.[4] It was determined that under certain conditions, the room which encloses the heating unit, or any room in the residence, can amplify a specific frequency of the noise from the heating unit, thereby making a normally quiet unit appear to be objectionally noisy. Remedial techniques for this are outlined in Reference 4.

PULSATION MECHANISM

From the results of the studies of furnace and burner variables, it has been possible to postulate a mechanism by which pulsations in oil-fired heating units are generated. This mechanism can perhaps best be understood if consideration is first given to the physical occurrences within the combustion chamber during pulsating and nonpulsating conditions.

A violently pulsating flame in an oil furnace will often appear to the unaided eye as a steady flame with some suggestion of a periodic flicker. Therefore, to study the actual movements of the flame front, it was necessary to use high-speed movies to "slow down" the flame movements. Many movies were taken of the flames of different burners, under different conditions of pulsation and nonpulsation. When the processed films were viewed at roughly 1/50 true speed, the cyclic events of the flame front became evident.

For nonpulsating conditions, the flame was initiated, even with the ignition off, at a constant distance downstream of the blast-tube exit, and combustion was continuous. Under pulsating conditions, the high-speed movies showed that combustion was not continuous, and, thus, that the position of the flame front was not constant.* Instead, the flame appeared to be initiated within the blast-tube exit or downstream from it and to expand rapidly as the gases passed downstream. Later in the cycle, luminous gases were seen in some furnace configurations to recirculate back to the blast-tube exit from regions downstream of the usual flame region to ignite the next batch of combustible mixture. In other cases, there was no visible recirculation of luminous gases; under these conditions it was surmised that hot, nonluminous gases were the means of periodic ignition.

Pressure measurements within the combustion chamber during pulsation indicated that there is an alternating acoustic pressure which is synchronized with the flame discontinuities. The amplitude of these pressure changes is such that it affects the capacity of the blower. When the pressure is low, a low pumping head is presented to the blower and an increasing flow rate of air results. With a constant fuel-flow rate, a large flame is pro-

* Prints from such a movie presented in a previous paper[1].

duced by the sudden excess supply of air into the temporary fuel-rich atmosphere of the combustion chamber. The hot gases from this flame build up the pressure within the combustion chamber, causing the pumping head of the blower to increase, and thus producing a corresponding decrease in air-flow rate. The decrease in air flow "starves" the flame for air and a smaller quantity of hot gases is produced. As a result, the pressure in the combustion chamber drops and the cycle is repeated. It is this alternating pressure that is called pulsation.

Of primary interest to this study was the determination of the manner in which the furnace and burner variables interact to sustain pulsations. As a first step in this determination, a theory on the driving of oscillations by combustion, based on Rayleigh's criterion,[6] was applied. In summary, Rayleigh hypothesized that a heat-driven oscillation could be sustained, if a pressure oscillation were kept in phase with a periodic rate of heat release, so that the maximum rate of heat release occurred when the acoustic pressure was above the average pressure. Note that two phenomena are involved: an acoustic pressure oscillation and a periodic rate of heat release. If either one of the phenomena could be eliminated, or if the phasing between them could be changed, then the pulsation could be eliminated. Each phenomenon will now be considered separately so that the furnace and burner variables which affect each of the phenomena can be understood better.

Pressure Oscillation – The characteristics of the pressure oscillation, which is considered in Rayleigh's criterion, are determined by a resonance condition in the furnace-burner system which is composed of a combustion chamber coupled with the burner on one end and a stack on the other. These components form, acoustically, a resonant system which has several natural frequencies. Although the flame within the combustion chamber produces noise over a wide range of frequencies, the furnace-burner system amplifies only those frequencies which are equal to the natural frequencies of the system. This flame noise is characterized as being a complex wave made up of a number of frequencies, all of which have low amplitudes, as contrasted to pulsation which is characterized by a single frequency of high amplitude.

Frequency measurements which were made of the noise from the experimental furnace during combustion conditions indicated that the frequency of the pulsation was usually about 32 cps. Acoustic measurements were made for "cold" conditions, without combustion, to determine the natural frequencies of the various components singularly and in combination. These natural frequencies were used to calculate the theoretical natural frequencies for the different burner-experimental furnace systems used during this program. Without combustion occurring, it was determined by calcu-

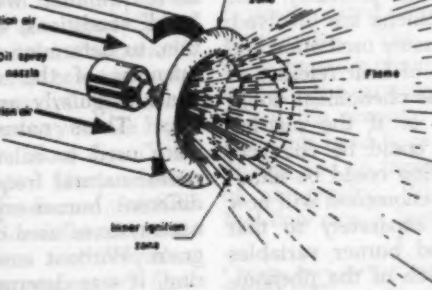
lation that the experimental furnace and a burner with a typical shutter-controlled blower inlet had two natural frequencies of practical interest. One of these frequencies was between 21 and 30 cps, and the other about 10 cps higher.

Periodic Heat Release – Periodic heat release results from periodic ignition of the combustible mixture. The factors which affect this periodic ignition are obviously the fuel-spray pattern, the air pattern, and the pattern of the recirculated hot combustion products. High-speed movies of the flame indicate that, during conditions of high-amplitude pulsation, the ignition of the combustible mixture occurs only in two doughnut-shaped regions of the combustion chamber and thus it is the quantity of fuel, air, and recirculated products within these regions which are important to the generation of pulsation rather than the average conditions of the entire combustion chamber.

Fig. 3 shows these ignition regions. An inner ignition zone was observed somewhat downstream to the blast-tube exit, near the burner axis, and within the fuel-spray cone. An outer ignition zone was considered to be concentric with the inner zone but closer to the blast-tube exit and outside of the main cone of fuel spray.

For some of the high-speed movies, a special technique was used by which a signal, indicative of the oscillating pressure amplitude in the combustion chamber, was superimposed on the film simultaneously with the exposure of the flame photographs. In this manner, the phasing between the ignition and the pressure amplitude could be determined. The ignition in the inner region occurred at about the time the oscillating acoustic pressure was at a minimum, and in the outer ignition region, by the proper combination of conditions, when the pressure

Fig. 3 Regions of periodic ignition



was at a maximum. However, it is not the ignition of the combustible mixture, but rather, the maximum rate of heat release which supplies the energy for driving the pulsation. There is a time lag from the occurrence of ignition to the attainment of this maximum rate of heat release. If this time lag is of the order of one-half of the period of the pulsation, it would appear from Rayleigh's criterion that pulsations would be driven from the inner ignition region when the proper mixture of fuel, air, and hot gases was available.

Based on the same value of time lag, a continual expansion of the flame from the outer ignition region would place an energy input in the wrong phase with the oscillating pressure to drive a pulsation. Consequently, in a system where both inner and outer ignition are occurring, pulsations which were driven by ignition and flame expansion from the inner region could actually be damped out by the outer ignition. But fuel is not available ordinarily in the outer region for such continual expansion.

It is possible, however, to drive a pulsation from the outer ignition region in the following manner. As observed in high-speed photographs for certain burner-furnace combinations, the expanding outer flame can be broken up and partially quenched by the reverse flow, and may even be pushed into the blast tube. Then the burning remnants are again projected into the combustible mixture as the gases rush back out, at above the average rate, to build up the pressure in the combustion chamber. These burning remnants serve as sources of ignition in the inner ignition region, and the flame expands at a rapid rate at the proper time. Thus, the heat needed for the periodic ignition can actually arise from either flame remnants or from hot gases.

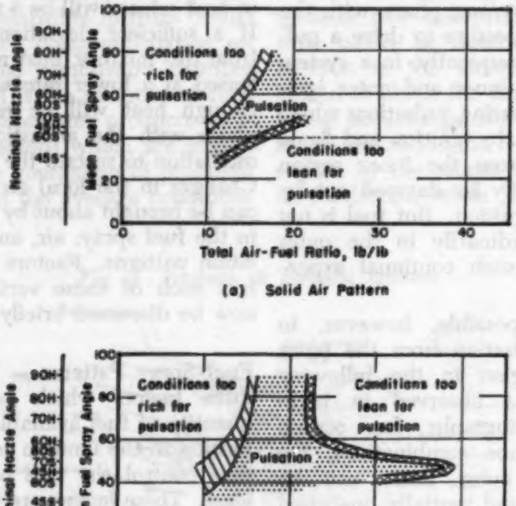
In order for ignition to occur in either the inner or outer ignition zone, a combustible mixture of vaporized fuel, air, and recirculated products must be present. The rate of heat release by the flame in these zones is controlled by the local air-fuel ratio in each zone. When the local air-fuel ratio is close to stoichiometric, the rate of heat release will be a maximum. If a sufficient deviation is made from this mixture, heat may be released at a lower rate so that not enough heat will be released in phase with the acoustic pressure oscillation to sustain the pulsation. Changes in the local air-fuel ratio can be brought about by variations in the fuel spray, air, and recirculation patterns. Factors which affect each of these variables will now be discussed briefly.

Fuel-Spray Patterns — There are three factors which control the quantity of fuel available for combustion in the ignition regions and thus control the rate of heat release. These factors are the (1) distribution of fuel droplet sizes, (2) heat available to vaporize, and (3) fuel-spray angle and pattern.

It has been shown by Binark and Ranz[6,7] that the distribution

of fuel-droplet sizes within a hollow spray exhausting into still air is partially determined by the induction of air by the fuel spray. Small droplets were found to be swept into the interior of the spray near the nozzle; these droplets were found near the spray axis. The larger droplets, which were swept into the spray interior further downstream from the nozzle, were found closer to the spray shell. The forced-air systems found in actual burners would affect these results, but the same relative segregation of droplet sizes would still be expected; thus, the fuel-droplet sizes within the inner ignition region would be conducive to a rapid evaporation and easy ignition. The ease of ignition is also affected by the amount of heat available. As has already been suggested, this heat comes from the recirculated hot-combustion products as well as from radiation from the flame and containing furnace walls. The fuel-spray angle and pattern also affect the availability of fuel in the ignition regions as has been summarized in the discussion accompanying Fig. 2.

Fig. 4 Nozzle-spray angles affect pulsations at various air-fuel ratios



(b) Hollow Air Pattern

Air Pattern – Different burners produce air patterns ranging between a solid and a hollow cone. Furthermore, the pattern tends to change with air-flow rate, especially the hollow air patterns. The shape of the pattern will determine how much air is made available in the ignition regions to burn the fuel.

Unlike the fuel-spray pattern, the air pattern is not constant during pulsation. The periodic variation in velocity will produce a corresponding variation in air-flow pattern. The flow pattern will form and then tend to collapse periodically, producing and shedding large doughnut-like vortices, and changing the quantity of air available in the ignition regions. During pulsation, this periodic variation in air pattern could set up a feed-back effect which would help to lock in the pulsation.

Recirculation Patterns – Recirculation, which is set up by the momentum of the air and fuel streams, is a natural phenomenon within the combustion chamber. Hot combustion products are returned to the flame region by the large recirculation loops. As with the air pattern, the presence of pulsations can affect the quantity of hot products which enter the ignition region to ignite the mixture.

Regions of Pulsation—The effects of the fuel spray, air, and recirculation patterns are interrelated so that in an actual burner system it is probably impossible to completely separate the effect of each variable on the presence of pulsation. However, during the course of the experimental studies conducted during this program, it was possible to define over-all conditions of pulsation and nonpulsation for the burners used even though the different variables were known to affect each other.

Fig. 4 is a map of pulsating and nonpulsating regions in terms of the mean fuel-spray angle, the total air-fuel ratio, and the air pattern as determined with the experimental furnace and burners described earlier. For each air pattern, it was found that pulsations occurred for certain ranges of air-fuel ratio and mean fuel-spray angle. In the case of the solid air pattern, a sufficient decrease in total air-fuel ratio and an increase in the mean fuel-spray angle caused the pulsations to cease, probably because the local air-fuel ratio in the ignition regions became low. The slope of the locus of points, where the pulsations ceased because of over-richness, indicates that when the mean fuel-spray angle increased, more fuel was caused to pass through the ignition regions. For the same air pattern, it is seen that increasing the total air-fuel ratio or decreasing the mean fuel-spray angle, or both, will cause pulsations to cease because the local air-fuel ratio in the ignition regions becomes too high.

It is noted that for the hollow air pattern, the general trends are the same as for the solid air pattern but the shape of the pulsation region is different. For the lean amplitude transitions, it is believed

that the mean fuel-spray angle swings through the ignition regions as the fuel spray is changed sufficiently.

The question might be asked why the shapes of the pulsation regions are different for the solid and hollow air patterns. One possible explanation is that the ignition regions shift somewhat with mixture ratio and air pattern. It seems reasonable that these regions would not have the same location at low air-flow rates at rich amplitude transitions as at high air-flow rates at lean amplitude transitions. One factor involved in such a shift of location of critical region is the variations in air velocity across the air pattern. In the case of the solid air cone, the velocity is highest on the burner axis and decreases with increasing radial distance. Changes in total air-flow rate cause equivalent changes in the air velocities throughout the pattern. The hollow air pattern, on the other hand, has a peak air velocity at some angle with the burner axis. Thus, with increasing radial distances from the axis, the air velocity is first found to increase and then to decrease in magnitude. It is reasonable to expect, therefore, that this complication in velocity profile could result in the more complex lean pulsation limit shown in Fig. 4 for the hollow pattern.

The data as presented in Fig. 4 apply only in a qualitative manner to other burner units. To obtain quantitative data strictly applicable to other units, it would be necessary to study the specific combinations in question.

CONCLUSION

Pulsations present in oil-fired heating units result from acoustical phenomena, even though the energy for driving the pulsations is derived from the combustion process. It is the acoustical system which determines the frequency, and to a large extent, the amplitude of the pulsations. Changes in furnace and burner variables which affect the combustion process also affect the acoustical system. The manner in which the system is affected, however, is not as evident as with the combustion process. It is this unknown effect of furnace and burner variables on the acoustical system which causes the seemingly inconsistent effectiveness of suppression techniques with different heating units. Thus, in this program, it became necessary to explain the mechanism of pulsation in terms of the important furnace and burner variables affecting the acoustical system. This has been accomplished to the extent that the variables important to the generation of pulsations have been delineated; these are the furnace volume, the fuel-spray pattern, the air pattern, and the pattern of the recirculated hot-combustion products. The complex nature of the physical handling of the fuel, air, and combustion products has, however, prevented a full determination of the effect of these variables on the acoustical system.

The detailed mechanism of pulsation has been shown to involve two independent phenomena: an acoustic pressure oscillation and a periodic rate of heat release. The

pressure oscillation has been shown to result from the acoustical behavior of the furnace. The periodic heat release is controlled by the periodic ignition of the combustible mixture in the furnace. This periodic ignition appears to depend on the local air-fuel ratio of the combustible mixture in the ignition regions downstream of the blast-tube exit. It is not fully known why these regions have dominance over ignition in any other region within the combustion chamber or how burner configuration affects the location of these regions.

Any suppression technique should affect either the acoustic-pressure oscillation or the periodic rate of heat release. Thus, such techniques as venting[8] the combustion chamber would be expected to affect the pressure oscillation by relieving the acoustic pressure within the furnace. Also, it should be possible to suppress pulsations by affecting the periodic rate of heat release. The problem is, that not enough is known about the location of the ignition regions or precisely what conditions are needed in these regions to prevent pulsations. Thus, it is not now possible to devise definitive techniques involving the fuel-spray pattern, air pattern, or recirculation pattern which will suppress pulsations in all circumstances.

This program has shown that additional information on the interrelationship of the fuel spray, air, and recirculation patterns is required before fully comprehensive pulsation-suppression techniques can be devised. Because of the multiplicity of furnace and burner designs in the industry, the evaluation techniques described should be directed to the specific cases of interest to individual manufacturers.

ACKNOWLEDGMENT

The able assistance of the members of the Pulsation Research Steering Subcommittee of the Technical Advisory Committee on Combustion of precedent ASHAE, in guiding the research and in providing necessary materials, is gratefully acknowledged. Special thanks are due the personnel of the Armstrong Furnace Company who constructed the experimental furnace used in this work.

REFERENCES

1. Putnam, A. A. and Dennis, W. R., "Pulsations in Residential Heating Equipment—Preliminary Results", ASHAE Transactions, Research Report No. 1596, Vol. 63, p. 153, 1957.
2. Putnam, A. A., Dennis, W. R., and Landry, B. A., Final Report from Battelle to U. S. Radiator Corporation, June 20, 1952.
3. Putnam, A. A., "Pulsations in Residential Gas Furnaces With Multiple-Port Burners", ASHAE Transactions, Research Report No. 1641, Vol. 64, p. 377, 1958.
4. Speich, C. F., Dennis, W. R, and Putnam, A. A., "Acoustic Coupling of Residential Furnaces With Their Surroundings", ASHAE Transactions, Research Report No. 1612, Vol. 63, p. 413, 1957.
5. Rayleigh, Lord, *Theory of Sound*, Dover Publications, Section 322g, 1945.
6. Binark, H. and Ranz, W. E., "Induced Air Flows in Fuel Sprays", ASME Paper No. 58-A-284, 1958.
7. Binark, H. and Ranz, W. E., "Measuring Drop Size of Hollow Cone Sprays" Industrial and Engineering Chemistry, Vol. 51, No. 5, p. 701, May, 1959.
8. Weeks, R. L., Clinkenbeard, W. L., and Soltis, J. D., "Clean Efficient Combustion of Heating Oil", Section VI, Paper 25, Fifth World Petroleum Congress, 1959.

DISCUSSION

R. C. WRIGHT, Cleveland, Ohio (Written): The statement, "In order for ignition to occur in either the inner or outer ignition zone, a combustible mixture of vaporized fuel and recirculated products must be present," appears to be not strictly in accordance with other observations reported in the paper. It was also indicated that burning flame remnants may accomplish reignition in the inner zone. In another part of the paper radiation from the flame and combustion chamber walls are indicated to have an effect upon the ease of ignition. Possibly, there are other as yet undiscovered mechanisms producing ignition.

The observation, "When the local air-fuel ratio is close to stoichiometric, the rate of heat release will be a maximum", may not have sufficient proof to justify accepting it as a definite conclusion. It might be that the rate of heat release is greatest when the mixture is fuel lean.

The classification of variables in Table I might be subject to critical analysis. For example, draft is listed as a variable having little or no effect upon the amplitude, yet it is also listed as having an effect upon the acoustical system, at least part of which is indicated to have an important effect on amplitude. Furnace configuration likewise is indicated to have little effect upon amplitude of pulsation, but is indicated to affect recirculation patterns. This is indicated to have an important effect on amplitude. Until more comprehensive knowledge is available, such classification may be misleading, especially from the standpoint of which is cause and which is effect.

There may be other yet undiscovered causes for some of the reported phenomena. Such positive reasons or explanations as presented might mislead or deter other investigations. That is not likely to be the case because future investigators who will solve some of these problems will probably be individuals accustomed to forming their own conclusions from the work of others. In the summary or conclusion it should be mentioned that there may be other important factors in addition to those of fuel spray pattern, air pattern or recirculation pattern. These are all important but the missing element may lie elsewhere.

AUTHOR PUTNAM: The statement in the text is "In order for ignition to occur in either the inner or outer ignition zone, a combustible mixture of vaporized fuel, air and recirculated products of combustion must be present in these local zones." Also, in all cases either remnants of flame or hot products are needed to ignite the fuel-air mixture. These conclusions resulted from noting that in all our observations of high-speed photographs of pulsating flames in residential-size oil-fired heating units, the pulsations appeared similar to those shown in Fig. 2 of Reference 1; the only differences were that they might be of a larger or smaller amplitude, and might or might not have outer ignition zone in addition to the inner zone.

The statement was made that the local air-fuel ratio (i.e., the air-fuel ratio in the ignition zone) had to be close to stoichiometric for the rate of heat release to be maximum. The reference here is to that part of the fuel that is vaporized or easily vaporized, following the thought of the preceding sentences. Only that part will burn in the short time of a part of a pulsating cycle and in a local region. The larger droplets will move on downstream, be burned later, and not contribute energy to driving pulsations. It is well known that the maximum burning velocity of a vaporized hydrocarbon-air mixture occurs close to and on the fuel-rich side of the stoichiometric mixture. The minimum flame thickness is also at about this mixture ratio, and thus the rate of heat release per unit volume is a maximum at about this value. It follows that where the local vaporized fuel-air ratio is near stoichiometric in the region critical to driving the pulsations, the maximum energy is available for driving the pulsations.

Table I attempted to segregate the major and minor factors which affected pulsation amplitudes. Major factors have a direct effect; minor factors must operate through the major factors. Thus, it would be expected that the effects of the minor factors would be smaller because of the expected inefficiencies of working through the major factors. Recirculation was considered a major factor; furnace shape a minor one since normally change in the shape will only affect a portion of the entire recirculation path, and may not even change the critical part of the path as far as pulsations are concerned. For instance, an extra row of bricks in the heating unit will change the furnace shape, and if these bricks are in the proper place, affect the recirculation zone slightly; thus, this change in configuration will have but a small effect on the pulsations.

Let us now consider draft. A change in draft without altering any other furnace variables can alter the frequency, the damping loss, and the acoustic radiation. But there is no large effect on any of these for normal changes in draft. A more important consideration is that in normal practice, a change in draft is not made without affecting the air-flow rate. This can have a large effect on amplitude, as shown by Fig. 4.

KENNETH A. MERZ, Torrington, Conn.: What was the magnitude of the pressure fluctuation within the firebox in the burner on which you experimented, both at the inception of pulsation and the maximum pressure fluctuations observed? Secondly, were you able to produce pulsations with no fan in the system but with constant air supply from a critical pressure ratio orifice?

AUTHOR PUTNAM: The peak amplitudes we

measured within the combustion chamber were as much as 140 db, which is equivalent to about 1.1 in. of H_2O peak pressure.

The answer to your second question is that pulsations can be produced with no fan in the system, and air supplied at a uniform rate through a critical flow orifice. In one study, we discharged from a critical flow orifice directly into the fan housing, to permit us to measure the air-flow rate. The fan was left in and operating. A normal amplitude of pulsation was obtained for that condition. On removing the fan and stuffing the fan housing with glass wool to even out the flow, we obtained one of the loudest pulsations in our experience.

We should point out that one of the early investigators, about 20 years ago, found that for his specific system the replacement of a fan by an air supply from a critical flow orifice eliminated pulsations. This observation has been generalized without justification. Our observations show that the concomitant changes in the acoustic system that usually occur when such a replacement is made are responsible either for the production or elimination of pulsations. Thus, whether a change from a fan to a high-pressure air supply will silence pulsations or not depends mainly on the changes in internal configuration of the air passages that are made to accommodate the change in air supply.

There is one occasion in which the change of a fan may be critical. If a system is just on the edge of pulsating, then it is possible that merely replacing the fan may cause pulsations, or stop them.

1747

No. 1747

Suppression of Pulsations in Oil-Fired Residential Heating Equipment

C. F. SPEICH A. A. PUTNAM

Elimination or suppression of pulsations in oil-fired residential heating equipment has been a goal of the oil-heating industry for many years. Trial-and-error approaches have resulted in many "cures" for pulsations which have been effective in specific instances. However, some cases of pulsation are found not curable by these techniques. Also, the mere availability of a "quick-fix" technique does not eliminate the cost of the service calls to alleviate pulsation problems. The real need has been for a more fundamental understanding of the causes of pulsations, so that antipulsating characteristics can be incorporated directly into the burner when it is manufactured. To obtain this more fundamental understanding, a research program was initiated at Battelle Memorial Institute in May, 1954, by the American Society of Heating and Air-Conditioning Engineers, Inc., the Oil-Heat Institute of America, Inc., and the American Gas Association.

A previous paper in this series has outlined a proposed mechanism for generation of pulsations in oil-fired heating equipment.[1] The mechanism has been shown to be complex, involving the proper phasing of an acoustic pressure oscillation and a periodic rate of heat release. Pressure oscillation arises from an acoustical resonance condition in the furnace-burner system, whereas the periodic rate of heat release arises from a periodic ignition of the combustible mixture. This ignition appears to be contingent on the local air-fuel ratio and presence of recirculated

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hot combustion products in a critical ignition region downstream of the blast-tube exit. The periodic heat release supplies energy to drive the pulsation while the acoustic pressure oscillation periodically sets up conditions for pulsed heat release.

This paper on suppression of pulsations in oil-fired heating equipment is the last of a series covering the results of the research program. It discusses suppression techniques which were investigated in this program or have been described in the literature.

DISCUSSION OF SUPPRESSION TECHNIQUES

Table I summarizes a number of different suppression techniques. Items have been grouped as affecting either the acoustic-pressure oscillation or the rate of heat release. Generally speaking, the frequency and amplitude of the pressure oscillation are dependent upon the design of the combustion chamber and flue passages, whereas the periodic heat-release rate is dependent upon the design and performance of the burner. Although phasing between the acoustic-pressure oscillations and the rate of heat release will also be affected by these various techniques, it probably is not possible to control this phasing to a degree necessary to suppress pulsations.

ACOUSTIC PRESSURE OSCILLATIONS

Generally, pulsations can be suppressed by either increasing the frequency or decreasing the amplitude of the acoustic pressure oscillations. As frequency of oscillation is equal to one of the natural frequencies of the furnace-burner unit, it can be increased by decreasing the furnace size. If the frequency were raised to a sufficiently high value, the fuel oil probably could not burn at a rapid enough rate to release heat at the proper time to sustain the pulsation. Also, high frequencies can be damped out more rapidly than can low frequencies by the internal configuration of the furnace. Sufficiently large frequency changes, however, are not normally practical because unit size is usually determined by such factors as heat transfer and volumetric heat-release rates, and, therefore, is not easily compromised for frequency changes.

Amplitude of oscillations can be decreased by increasing damping losses of the furnace-burner system. Through damping, energy is removed from the oscillating system by acoustic radiation, frictional losses, or flow resistance. Acoustical radiation can occur when noise leaves the system through an opening in the furnace. To a listener outside the heating unit, the opening may appear to be a noise source. On the other hand, if this loss in acoustic energy prevents pulsation, a net gain in reducing the total noise output of the furnace-burner unit is achieved.

Frictional losses occur in small openings where viscosity effects of fluid predominate. Sanders and Lawrie[2] have presented an equation for determining the magnitude

TABLE 1—SUMMARY OF PULSATION SUPPRESSION TECHNIQUES

Techniques	Remarks
Changes Affecting the Acoustic Pressure Oscillations	
Venting of the Combustion Chamber	By relieving acoustic pressure in the combustion chamber, large decreases in the excess air can be made without incurring pulsations.
Acoustic Filers	Because pulsation frequencies are low (15 to 40, cps), large physical volumes are usually required to suppress pulsations. These large sizes make this method generally impractical.
Changes Affecting the Periodic Heat Release	
Fuel Spray Pattern	This common technique can suppress pulsations if large enough changes are made in spray angle or pattern.
Burner Air Pattern	Not enough is known about this technique to make quantitative predictions of its effectiveness. Present changes are made on a trial-and-error basis.
High Δp Across Choke	Pressure drops of the order of 2 in. of H_2O may be required to isolate the burner acoustically from the furnace by this technique.
Fine-Scale Turbulence	No quantitative information is available by which good mixing can be predicted with minimum noise generation.
Fan Characteristics	Pulsation suppression, which is usually accredited to fan changes, is more likely due to simultaneous changes in the acoustical system.
Recirculation	Some recirculation-type burners have been reported to operate quietly after recirculation has been fully established. The effect of the recirculation on the pulsation and combustion noise amplitudes has not been established, however.
Flame Stabilization	By affecting the velocity gradients of air, fuel, and recirculation patterns, steady combustion can be achieved over wide ranges of air-fuel ratios.
Miscellaneous	
Room Acoustics	If the enclosing room becomes active in amplification of incipient pulsation or combustion noise from the furnace, a relocation of the heating unit or partitioning in the room may be necessary.

of this loss. They also give an equation for predicting the magnitude of an acoustic resistance due to fluid flow in the burner tube or flue passages. This last resistance can provide some degree of acoustic isolation among the various components of the furnace-burner system.

Practical application of these damping techniques to oil-burning heating equipment has consisted of

intentional venting of the combustion chamber, and use of acoustic filters in flue-gas-handling parts of the furnace.

Venting the combustion chamber—To paraphrase an article in the trade literature,[3] to stop pulsation—drill one hole in the front of the furnace. If that doesn't work—drill two holes! Based on this example alone, one might place little confidence in the technique of suppressing pulsations by venting. On the other hand, venting has been used successfully by the industry as a means of preventing pulsation.

Glendenning[4] has reported the pulsation suppression effect of a venting device consisting of a series of ports concentric with the blast tube; a shutter allowed the vent area to be varied. Glendenning suggested that admission of secondary air through these ports prevented the formation of low-pressure surrounding the flame region. However, in view of additional information now available, it seems more likely that acoustic effects were responsible for the success of the device.

Recent work by Weeks and others[5] has shown that an optimum venting area exists for a given furnace-burner unit to achieve suppression of pulsations with a minimum loss in efficiency or production of smoke. Using their technique, venting is achieved by placing a series of fixed-area ports in a ring concentric with the blast tube of the burner. The axes of the ports are parallel to the burner axis. In a succeeding paper,[6] Sage and Schroeder have indicated that beneficial effects of venting are derived from an acoustical effect rather than the net influx of secondary air. Their conclusion was based on an experiment in which the percentage of excess air at which pulsation commenced (comparable to the "lean transition point" defined in a previous paper of this series[1]) of venting and nonventing. A special vent plate which had several holes manifolded to an auxiliary air supply was installed. Equal amounts of secondary air were fed into the combustion chamber during both venting and nonventing conditions. The percentage of excess air at which pulsations were eliminated was lower for vented than for nonvented combustion chambers. On the other hand, supplying additional quantities of secondary air through these ports produced an increase in the percentage of excess air at which pulsations ceased. Thus, the presence of secondary air negated the beneficial effects of venting.

During the experimental phase of the Battelle program, an observation was made of the effect of unintentional combustion chamber venting by cracks and leaks in the chamber wall or the joint between the burner and furnace. It was found that sealing these cracks caused a large increase in the pulsation amplitude from one burner, characterized by a solid air pattern, but the same care in sealing had little effect on noise from another burner, characterized by a hollow air pattern. In an attempt to explain this large effect of leaks on

the pulsation amplitude from the first burner, the possibility of secondary air influx, changes in the system's natural frequency, or increases in the damping losses in the furnace-burner system were considered. Insufficient data were available to ascertain which of these was the real cause of the observed amplitude change.

In summary, it can be said that the effect of venting the combustion chamber is to retard build-up of an acoustic oscillation, and thus, to allow the burner to operate at lower percentages of excess air than would be possible with the unvented system. The extent of reduction in percentage of excess air which is possible without pulsation, objectionable smoke, or large decreases in efficiency will depend on the burner-furnace unit.

Acoustic filters—Use of acoustic filters has been suggested as one means of achieving an apparent suppression of pulsations and combustion noise. This technique permits generation of noise, but depends on acoustical devices to attenuate the noise before it leaves the heating system.

"Handbook of Oil Burning" illustrates three techniques by which the flue system can be modified to reduce the amplification of combustion noise and pulsations.

These techniques can be interpreted as being (1) use of a branch on the smoke pipe to increase acoustic radiation losses of the heating system, (2) changing effective length and thus, resonant frequency of the chimney, and (3) use of a high-pass acoustic filter assembly in the smoke pipe. The last item attenuates low frequencies of combustion noise, but it allows high frequencies to pass unaffected.

Putnam and Dennis have described the use of acoustic dampers for oscillation suppression in combustion systems[8]. In particular, they discuss the application of quarter-wave tubes, Helmholtz resonators, and venting orifices as means of suppressing oscillations. Quarter-wave tubes, Helmholtz resonators and large diameter orifices which vented the combustion chamber to the atmosphere were found to be effective over a wide range of locations in the chamber. Small diameter orifices were found to be most effective at or near regions of maximum acoustic pressure.

Major drawback to use of acoustic filters is that they must be physically large to attenuate properly low frequency pulsation generated by oil-fired equipment. If size were no problem, however, they would be quite effective for attenuating single frequency noises, such as pulsation.

PERIODIC HEAT RELEASE

As has been suggested already, periodic variation in the rate of heat release results from a periodic ignition of the combustible mixture. This ignition, in turn, is believed to depend upon the quantities of vaporized fuel, air, and recirculated hot combustion products available in a proposed critical ignition region downstream of the blast tube exit.

To prevent pulsations, what is probably required is that the local air-fuel ratio and quantity of recirculated products in the critical ignition region be such that insufficient heat is released in phase with the oscillating pressure to sustain the pulsation. Thus, to affect the pulsation amplitude by changing the periodicity of the rate of heat release, control must be exercised over the fuel, air, and recirculation patterns. Unfortunately, in an oil burner, control exercised over these parameters appears to decrease in this same order. That is, fuel-spray angle, pattern, and quantity of fuel are regulated quite closely. This control arises primarily from high pressure drop across the nozzle which imparts a high momentum to the fuel droplets; thus, the fuel flow is isolated from pressure fluctuations in the combustion chamber. The pressure drop across the burner choke is relatively small; thus, the air-flow rate and pattern are partially controlled and there is only a partial isolation of the air supply from pressure fluctuations in the combustion chamber. In conventional furnace-burner systems, there is little or no control over the recirculated hot combustion products since they are located in the combustion chamber and are, therefore, at the mercy of the pressure fluctuations.

Fuel-spray pattern — For many years, servicemen have used, with varying degrees of success, the technique of changing the fuel-nozzle angle or spray pattern as a means of preventing pulsations. Experience has shown, for instance, that hollow-cone sprays usually give more freedom from pulsation than do solid cone sprays.[9] One difficulty with this technique has been that, with any particular burner unit, a sufficient change in

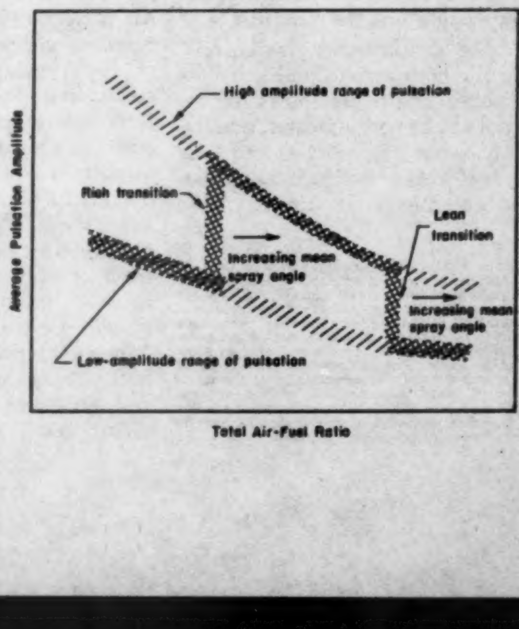

Fig. 1 Low- and high-amplitude ranges of pulsation as related to a solid air pattern and typical amplitude observations for one nozzle

angle or shift in fuel pattern to stop pulsations often results in smoke, odor, or poor efficiency.

As part of the experimental phase of this research program[1, 10], a systematic study was made of the effect of nozzle spray angle and pattern on the amplitude of pulsation.

Fig. 1 is a sketch showing the results of these fuel-spray studies. Generally, noise amplitude from the heating unit was found to decrease with increasing air/fuel ratio. However, with some spray types and angles, sudden increases from low-amplitude noise to high-amplitude were found to accompany an increase in air-fuel ratio. For still other spray types and angles, a sudden transition from a high amplitude to a low one was noted with an increase in air/fuel ratio. Studies of a large number of different angles and patterns showed that the points where these amplitude transitions occurred correlated quite well with the mean spray angle* of the nozzle.

Fig. 2 illustrates this correlation for both a solid and a hollow air cone. The figure shows regions of pulsations and indicates, qualitatively, what spray angles and total air/fuel ratios produce nonpulsation conditions. It can be noted from Fig. 2b that burners incorporating nozzles with large mean fuel-spray angles are generally free from pulsations at normally used air/fuel ratios. As discussed in detail in a previous paper,[1] this freedom from pulsations has been attributed to the increase in air/fuel ratio.

It should be noted that, for purposes of establishing limits for research analysis, the range of conditions examined in the summary of Fig. 2 was broader than the practical operating range of conventional burners. As an additional sophistication, another curve could be placed on the figure representing the limit for objectionable smoke, thus following the example of Weeks, et al.[5] Manufacturers who analyze their units in this general manner could interpret the results for the field serviceman and thus recommend nozzle types which would operate successfully in the desired ranges.

Air pattern—This modification represents a generally less used means of suppressing pulsations. This stems from the fact that air patterns are quite often fixed for a particular burner and thus changes in pattern would represent modifications in blast tube parts. However, there are combustion heads for which air patterns can be varied from sharply hollow to solid air cones. The literature suggests that a solid air cone has a greater resistance to pulsation than does a hollow air cone.[11] This increase in resistance can also be seen in Fig. 2 for small mean spray angles.

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* In the analysis of the fuel-spray pattern, it was found convenient to replace the nominal designation of fuel sprays, namely, spray angle and pattern, with a hypothetical spray-pattern model. This model defined all sprays, regardless of angle or pattern, as a thin, hollow spray in which all of the fuel was concentrated at some effective angle. The model's effective angle of spray was defined, mathematically, as the mean angle of mass concentration of the original spray and was calculated in a manner similar to the determination of the center of gravity for a linear distribution of masses.

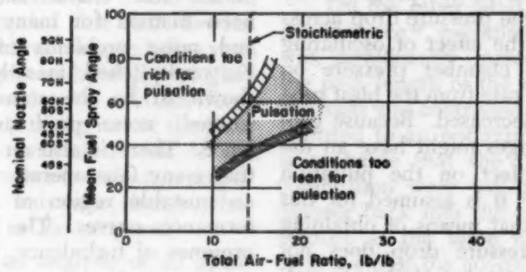
Presence of a positive flow of air in the center of the air cone may have the effect of increasing the acoustical resistance of the blast tube.

Little definitive information is available in the literature on air patterns produced by oil burners. Manufacturers are often aware of the general air pattern produced by their burners but seldom determine, quantitatively, air pattern, either when exhausing in the open, or when the burner is inserted into a furnace. As an additional complication, periodic pressure changes within the combustion chamber will produce corresponding changes in flow patterns; this periodic change can feed back into the oscillating systems and help "lock in" the pulsation.[1] The need, therefore, exists for additional knowledge of air patterns under pulsating as well as under normal flow conditions in a furnace, possibly as determined by the use of flow tracers and high-speed movies in a modified furnace. This information would also be of great value in optimizing fuel and air mixing.

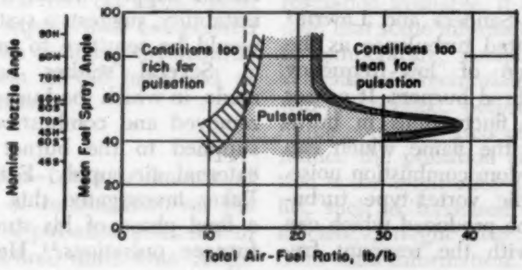
High-pressure drop across choke—As a means of increasing acoustic

Fig. 2 Nozzle-spray angles affect pulsations at various air-fuel ratios

A — Solid air pattern



B — Hollow air pattern



resistance in the blast tube, Sanders and Lawrie[2] have suggested the use of a large pressure drop across the choke. Measurements made as a part of this research program indicated that pressure drops across the choke of a burner under nonpulsating conditions were of the order of 0.4 in. of H_2O. On the other hand, sound-pressure levels of the order of 140 db have been measured within the combustion chamber during pulsating conditions. In terms of an instantaneous pressure, this sound-pressure level represents a swing in peak pressure of about 1.1 in. of H_2O above and below the ambient pressure. This high oscillating pressure explains why back flow into the blast tube has been observed in some high-speed movies in this program. By increasing the pressure drop across the choke, the effect of oscillating combustion chamber pressure on the air-flow rate from the blast tube would be decreased. Because geometric changes might have an unexpected effect on the pulsation amplitude,[12] it is assumed for this discussion that means of obtaining the high-pressure drop does not alter the geometry of the air pattern.

Turbulence—Sanders and Lawrie[2] have accredited turbulence as the major source of low-frequency noise in oil-fired burners. It causes the random fluctuations in burning rate of the flame which can produce random combustion noise. Also, periodic vortex-type turbulences can be produced which can "lock in" with the resonant frequency of the furnace-burner system and produce a periodic air flow and, therefore, pulsations. Sanders and Lawrie suggest the generation of small-scale, instead of large-scale, turbulence by the burner. This small-scale turbulence would permit adequate mixing of fuel and air but would die out rapidly enough so as to provide smoother combustion. Unfortunately, at present it is not known what scale of turbulence is needed to produce this adequate mixing with a minimum in the production of fluctuations in the burning rate of the flame. Means of producing small-scale turbulences are outlined in the appendix of their paper.[2]

Fan Characteristics—In the past, burner-fan characteristics have been blamed for many pulsation and noise problems of burners. Forward-pitched fans have been shown to be the source of unwanted, noise-producing turbulence.[2] There is also an indication that many fans operate near or in an unstable region of their performance curves. The combined presence of turbulence, which can produce random fluctuations in the heat-release rate of the flame, and an air supply, which borders on instability, suggests a system which would be sensitive to pulsations.

Several studies have been made, in which the burner fan was removed and combustion air was supplied to the burner from an external air supply. For instance, Baker investigated this system as a final phase of his study of oil-furnace pulsations.[13] He supplied

compressed air directly into the blast tube with the burner fan sealed off from the blast tube. Comparisons were made of the pulsation amplitudes under this condition of air supply with normal operating conditions at the same percentages of CO_2. In all cases, the amplitude of pulsation was found to decrease substantially. Using a technique similar to Baker's, Putnam and Dennis[12] conducted several tests on a series of standard blast-tube boiler combinations. In most instances, replacement of fan air by compressed air reduced the amplitude of pulsation to about half. Similar studies were also made in the present program. In one case, compressed air was supplied to the burner for combustion, and the noise output of the system was determnied first with the fan in place and running, and then with the fan replaced by a pad of glass wool to produce a uniform flow. Replacement of the fan by the glass wool pad resulted in one of the largest increases in pulsation amplitude, with only one change, that was noted in the program.

From an analysis of all available results, it was concluded that the change in pulsation amplitude does not stem from the replacement of the air supply with compressed air, but from changing the acoustic configuration at the inlet end of the blast tube.

Recirculation—Little is known of the effect of recirculation on the amplitude of pulsation and random noise in oil-fired units. One design of controlled recirculation-type burner has been reported to operate exceptionally quietly after recirculation has been established.[14] This burner has a configuration aimed at augmenting the prevaporization of the fuel. Another feature of the burner is that the residence time of the fuel in the reaction zone of the combustion chamber is longer, and the heat can be released more slowly than in a conventional unit. The capability for burning to occur anywhere within a large region of the combustion chamber, plus random timing of the burning, would appear to preclude any chance of a coupling between the rate of heat release and an acoustic pressure oscillation within the combustion chamber.

On the other hand, in at least one instance[12], pulsation amplitude was significantly increased when the path length of the recirculated products was controlled closely and adjusted to an optimum length. It is probable that the products of combustion were led in a pulsed manner back into the combustible mixture at the same time during each cycle of the acoustic pressure oscillation.

From the small amount of information available, it is quite evident that some information relating to temperature, composition, and velocities of recirculated products will be needed, before this technique could be used effectively to reduce the pulsation and combustion noise amplitudes. Other benefits, such as increased combustion efficiency, might also be derived from such information.

Flame Stabilization—It has been observed that the flame front of a gun-type burner does not always remain fixed with respect to time or space[4]. Instead, it may move randomly about, as during non-pulsating conditions, or in a periodic manner, as during pulsating conditions. One means of providing a fixed position for the flame front is to use a flameholder.

The mechanism of flame stabilization is not fully agreed upon. However, flameholders are known to perform at least two functions. First, they change the local stream velocity in such a way as to better match this velocity with the burning velocity of the fuel-air mixture. Secondly, a flameholder induces recirculation in a fixed region in which hot gases are retained and supplied into a critical region of the oncoming stream. Thus, with a flameholder, proper conditions for ignition and combustion of the fuel-air mixture are provided artifically in a fixed position. This is in contrast to a gun-type burner without a flameholder in which the air, fuel, and recirculated products must find an aerodynamically formed region where ignition and combustion can be sustained. This region will tend to float with changes in air or fuel flow patterns.

Because it is situated in the fuel-air stream, a flameholder is subject to deposit problems. These do not occur, normally, with continuously operating burners, but they have been a serious problem with intermittently fired burners such as are in residential use. Banscher[16] has patented a burner-flameholder combination for which he states that deposits will not occur because the combustion air sweeps over the flameholder properly.

Another method of flameholding uses reversed jets.[17] Although not likely to be practical for residential oil burners, the concept of reverse jets does show how proper use of air or air-fuel jets can provide flame stability and avoid the deposit problem inherent with some solid flameholders.

ROOM ACOUSTICS

As has been suggested by Speich, et al.[18] and by Sanders and Lawrie,[2] the room in which a heating unit is located can sometimes amplify the incipient pulsation from the unit to an objectionable level. This condition will exist when a dimension of the room is equal to one-half (or mulitple thereof) the wavelength of the principal frequency of pulsation, and when the heating unit is in proper location within the room. The only practical solutions to the problem of room acoustics appear to be to change the location of the heating unit, to use auxiliary walls or baffles to change the natural room frequency, or to replace the furnace by one with a different frequency. Proper location for auxiliary walls or the heating unit has been suggested.[18] Acoustic treating material, such as fibrous materials, are not effective noise suppressors for the low frequencies involved in residential oil-fired heating units.

CONCLUSION

The following comments are made, relating to pulsation suppression techniques for oil-fired residential heating equipment:

1. Venting appears to be an attractive "quick-fix" type technique which can be applied to most heating units. Possible escape of smoke, odor, flame, and random combustion noise from the vent must be considered in any design, and thus, may represent a drawback to this technique in some cases.

2. The value of nozzle changing can be enhanced, if systematic studies are made of a given furnace-burner unit, in order to delineate the regions of pulsation, as has been shown in Fig. 2.

3. A promising technique for suppressing pulsations is air-pattern modification. However, extensive information would be needed to provide basically applicable information. If better control of the air pattern were possible, other combustion aspects than pulsation control could also be benefited.

4. The air-handling system of gun-type burners can be made more stable by incorporating a high pressure drop across the choke, producing a fine scale turbulence, and by using a fan with a performance curve more suitable for oil-burner application.

5. Controlled recirculation appears to have a beneficial effect in suppressing generation of combustion noise and pulsation.

6. Use of flameholders appears to have great promise for control of pulsation, if they can be made inexpensively, and if they can be designed to minimize deposit problems.

7. Suppressing pulsations by using acoustic filters or by increasing the frequency of acoustic pressure oscillations does not appear to be practical for oil-fired residential heating units.

ACKNOWLEDGMENT

The able assistance of members of the Pulsation Research Steering Committee of the ASHAE Technical Advisory Committee on Combustion in guiding the research and in providing necessary information and materials is gratefully acknowledged.

REFERENCES

1. Putnam, A. A. and Speich, C. F., "Combustion-Driven Pulsations in Oil-Fired Residential Heating Equipment", to be published.

2. Sanders, G. J. and Lawrie, W. E., "Low Frequency Combustion Noise in Oil-Burning Equipment", Heating, Piping and Air Conditioning, Vol. 30 (10), p 127, October, 1958, and *ASHRAE Transactions*, Vol. 65, p 181, 1959.

3. Anonymous, "To Reduce Pulsations", Fuel Oil Journal-Oil Heating and Air Conditioning, Vol. 18 (10), p 16, April, 1940.

4. Glendenning, E. B., "Shell Anti-Pulsator", Report No. PAL-107, Shell Oil Company, Sewaren, N. J., June 10, 1948.

5. Weeks, R. L., Clinkerbead, W. C., and Soltis, J. O., "To Modernize the Old Installation", Fuel Oil News, Vol. 25 (2), p 22, February, 1960.

6. Sage, R. W. and Schroeder, H. F., "Noise Suppression in Oil Burners", ASHRAE Journal, Vol 2 (10), p 53, October 1960.

7. Faust, F. H. and Kaufman, G. T., Handbook of Oil Burning, Oil Institute of America, p 459, 1951.

8. Putnam, A. A. and Dennis, W. R., "Suppression of Burner Oscillations by Acoustic Dampers", ASME Transactions, p 875, 1955.

9. Olson, E. O., "Application of Fuel Nozzles in High-Pressure Domestic Oil Burners", presented at the Third Symposium of the Technical Division, Seattle, Washington, Oil-Heat Institute of America, Inc., April 30, 1959.

10. Putnam, A. A. and Dennis, W. R.,

"Pulsations in Residential Heating Equipment", ASHAE Transactions, Research Report No. 1596, Vol. 63, p 153, 1957.

11. Dennis, R. L., "Pulsation and Compression Factors", Fuel Oil News, Vol. 20 (12), p 16, December, 1955.

12. Putnam, A. A., Dennis, W. R., and Landry, B. A., "Final Report from Battelle to U. S. Radiator Corporation, June 30, 1952.

13. Baker, J. O., "Determination of the Pulsating Characteristics of High-Pressure Oil Burners", Masters Thesis to Iowa State College, 1941.

14. Sullivan, W. A. and Ventres, L. H., "A New Combustion Process", Fuel Oil News, Vol. 24 (6), p 33, June, 1959.

15. Banscher, F., "Burner for Liquid Fuel", U. S. Patent No. 2646109, July 21, 1953.

16. Cambel, A. B., "A Review of Flame Stabilization by Means of Gaseous Jets", Combustion and Propulsion — Third AGARD Colloquium, Pergamon Press, p 541, 1958.

17. Speich, C. F., Dennis, W. R., and Putnam, A. A., "Acoustic Coupling of Residential Furnaces With Their Surroundings", ASHAE Transactions, Research Report No. 1612, Vol. 63, p 413, 1957.

DISCUSSION

R. C. WRIGHT, Cleveland, Ohio (Written): This paper makes numerous references to "acoustic" pressure oscillations. Acoustical pulsations are not necessarily the only pulsations having a detrimental effect upon combustion. Pulsations of a frequency not audible have been observed and should be considered. Strictly speaking, the pressure oscillations generally referred to as pulsations are not necessarily always acoustical.

It would be advantageous to know whether the range between the rich transition and lean transition points illustrated in Fig. 1 is sufficiently definite to illustrate more specifically by showing the actual air fuel ratio in pounds per pound as has been done in Figs. 2A and 2B.

The hollow air pattern analysis shown in Fig. 2B may be misleading. This indicates a wide pulsation area in the general range of 20 to 50% excess air, which is generally the most desirable operating range. This might lead to the conclusion that a hollow air pattern is undesirable. Both Figs. 2A and 2B illustrate air patterns, yet apparently the mean fuel spray angle for both solid and hollow fuel spray patterns is shown in the same air pattern analysis.

References in the paper indicate that "hollow cone sprays usually give more freedom from pulsation than do solid cone sprays". If the fuel spray is an important factor, it would seem desirable to isolate this in the presentation of the effects of air spray pattern.

AUTHOR SPEICH: The first statement, dealing with acoustic pressure oscillations, is a matter of semantics. Perhaps, the term that Mr. Wright would prefer here is "audible" rather than acoustic. We maintain that the combustion-driven oscillations are dependent upon the acoustic characteristics of the furnace-burner system. This is not to say, however, that the oscillations or pulsations will always be audible. Specific examples can be given of furnaces and small blowers which have pulsating frequencies as low as 12 cycles per second, which is below the accepted limit for audibility. (cf. Fig. 2 of Reference 17.)

With regard to more quantitative information on Fig. 1, which shows rich and lean transition points, it was intended to make this qualitative rather than quantitative since other furnace-burner systems would have transition points at different air-fuel ratios for the regions of high-amplitude pulsation or low-amplitude noise. Also, other furnace-burner systems might not show as large an amplitude change as the 20 to 1 change noted in some of our studies.

In operating a furnace, it is normal not to operate over as wide a range of air-fuel ratios as was done in our studies. Thus, only a certain portion of the curve might be of interest, such as the high-amplitude pulsation or the low-amplitude noise regions or one of the transitions. The presentation of our studies was purposely made more qualitative so as to indicate the pulsation characteristics of a generalized system one should look for. The numbers for a specific case would have to be determined by individual manufacturers.

It was shown that pulsations would occur anywhere in the general range of 20 to 50% of excess air for our specific unit. With a different furnace-burner system, different values would be expected.

Since both fuel and air are required for combustion, it is not possible to separate completely the effects of the injection of each into the combustion chamber on the pulsating characteristics of a furnace-burner system. Thus, to obtain the data used for Figs. 2a and 2b, a consistent set of fuel-spray nozzles was used for each of two different air patterns in order to separate the effect of each pattern on the pulsation characteristics of the combustion system. For our unit, the implication of Fig. 2 is that narrow nominal spray angles should be used with solid air patterns, and wide nominal spray angles used with hollow air patterns to minimize the range of pulsations. It so happens that all the wide nominal spray angles

were obtained from hollow cone nozzles, and this may explain some of the results reported in the literature. But our data do not indicate that it is the hollowness of the spray that is of primary importance but the nominal angle.

1748

No. 1748

Noise Suppression in Oil Burners

R. W. SAGE
Associate ASHRAE

H. F. SCHROEDER

Importance of noise to the purchaser of heating and air conditioning equipment has been amply demonstrated. ASHRAE has devoted considerable effort toward understanding and solving the problems associated with noise. Much of this effort has been directed at the problems of mechanical noise such as compressor and blower operation, duct noise, insulation, etc. In oil-fired furnaces and boilers, however, much of the noise originates in the oil burner itself.

Burner noise is created by both the flame and the mechanical components of the burner. However, as shown in a previous paper,[1] the flame is the major source of oil burner noise. Hence, an experimental study of oil burner flame noise was carried out to determine the mechanisms by which the noise was produced. The objective was to develop an effective means of reducing noise level by simple adjustments to existing equipment.

Two types of flame noises were examined: high level throbbing noise or "pulsation," and low level normal noise or "combustion roar." The former type is not prevalent in most burners, but when it does exist, severe complaints are received from homeowners. The annoyance of oil burner noise can be compared with common household appliances.[2,3] For example, a pulsating oil burner is as annoying as a vacuum cleaner, a nonpulsing burner approximates a window air conditioner and the mechanical noise from a quiet oil burner is about the same as a new refrigerator.

A bar graph comparing these annoyance levels is shown in Fig. 1. This comparison assumes that the listener is standing next to the oil burner. However, if the burner is in the basement and the listener is in the living room, the burner

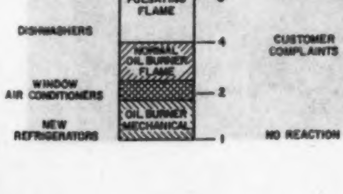
R. W. Sage ad H. F. Schroeder are with the Process Research Div, Esso Research and Engineering Company. This paper has been prepared for presentation at the ASHRAE Semiannual Meeting, Chicago, Ill., February 13-16, 1961.

sound level would be reduced at least 10 db.[4] This means that a pulsing flame in the basement would be as annoying as a dishwasher when listening to it in the living room. However, even these lower annoyance levels are still too high and should be reduced.

Noise Levels Measured With Conventional Equipment — A typical furnace-burner combination was chosen to examine both types of flame noise levels. Two conventional warm air furnaces with output ratings of 72,000 and 90,000 Btu/hr were used. They are representative of the smaller sized furnace which is gaining popularity in new homes today. The oil burners tested in these furnaces were also typical of those found in the field. A commercially available high pressure gun burner was used to study normal combustion noise. To study pulsation flame noise, a typical low pressure air atomizing burner was used.

Standard sound instruments were used to determine the sound spectrums of the noise from the oil burners. Burner noise is composed of sound waves of all frequencies in the audible spectrum (20-20,000 cycles/sec). However, in most test work, it is not necessary to analyze the spectrum for sound pressure level as a continuous function of frequency; rather, the sound level is measured in "octave bands." These octave bands have the same frequency limits as octaves on the piano keyboard. The sound pressure level gradient obtained by this method gives a good approximation of the entire spectrum and was used in this analysis of burner noise.

Fig. 1 Relative loudness of oil burner noises



Sound instruments used in this program gave results reproducible within one db. Under normal conditions the human ear can barely detect this one-db change in sound pressure level. A commercially available sound level meter was used for detection, and sound measurements in the octave bands were made with a standard octave band frequency analyzer. These instruments were checked and calibrated daily. They were always located in the same position relative to the furnace and this location was chosen as the point at which the sound pressure level was most critical. The microphone orientation is shown in Fig. 2. A cathode-ray oscilloscope and oscillator were used to determine the pulsing frequencies.

All the sound measurements were made in a remotely located sound laboratory. The background noise of this laboratory was extremely low and equivalent to the noise level of a radio broadcasting studio. The sound pressure level

gradient obtained was at least 30 db below the total burner noise in any given octave band. Since this background noise level was much less than the burner noise, it could not contribute to the total noise level.

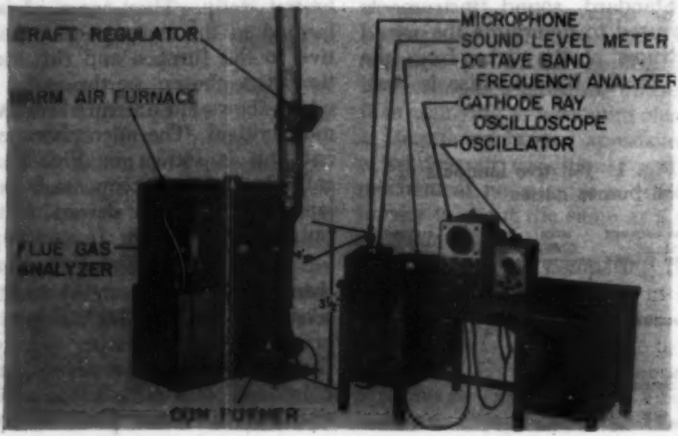
PULSING FLAME NOISE

Pulsation Studied With Aid of High Speed Motion Pictures—The loudest and most annoying noise found in domestic oil burners is pulsation. This phenomenon does not occur in all oil burners. But when it does, it dominates all other noise and becomes a focal point for serious complaints. Therefore, the first phase of our noise study was concerned with pulsation and how it could be controlled. Our program consisted of determining:

1. How pulsations originate and are perpetuated.
2. What effect burner and furnace variables have.
3. What simple, practical device can be built to eliminate pulsation.

A pulsing oil burner flame in the warm air furnace was photographed at 3000 frames/sec. These pictures showed that the flame was flickering "on" and "off" in the combustion chamber. The flame was not anchored in the normal position near the burner blast tube, but periodically traversed the combustion chamber and disintegrated against the rear wall. At the same time a new flame front was forming which followed the same procedure. At all parts of the cycle, some flame always existed in the combustion chamber. This periodic flame extinction coincided with the predominant 30 cycle sound wave measured on the oscilloscope, indicating that a definite relationship existed between pulsation noise and flame extinction. A

Fig. 2 Laboratory noise test equipment



typical frame from the high speed movies is shown in Fig. 3.

These high speed motion picture studies and supplementary tests enabled a pulsation mechanism to be postulated. The energy of the sound waves generated by the combustion process can be expressed by the classical formula:

$$I = \frac{P^2}{dC}$$

where:

I = sound intensity or energy per unit time flowing through a unit area

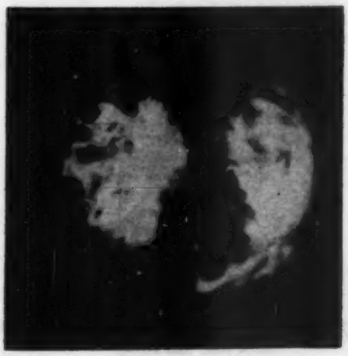
P = root mean square value of instantaneous pressure over the given time interval

d = density of combustion gases

C = velocity of sound in combustion gases

It was postulated that pulsation occurs when the energy level of the sound waves in the furnace is sufficient to retard significantly the flow of combustion air through the burner blast tube. The magnitude of this energy is dependent largely on the pressure level or amplitude of the waves as shown by the equation.

Fig. 3 Pulsing flame. Top view of combustion chamber. Air and oil enter on left



When resonant conditions exist between the flame and furnace, high amplitude sound waves of this resonant frequency build up to the energy level required to retard the flow of combustion air. A rich fuel-air mixture therefore is produced momentarily which cannot support combustion, at which time the flame "goes out." This produces a low pressure zone and results in a surge of air into the combustion chamber. Re-ignition occurs and the flame is re-established to complete the cycle. This periodic flame extinction and re-ignition occurs about 30 times per sec and can be observed easily by high speed motion pictures. To the naked eye, however, the flame appears to fill the chamber completely at all times. This mechanism appears to be in agreement with the findings of Sanders and Lawrie[5] and Putnam and Dennis.[6]

This mechanism opened two avenues of approach to the problem:

(A) Make the flame less sensitive to changes in air delivery.

(B) Dampen the acoustic system and thereby minimize changes in air delivery.

Flame Stabilization Prevents Pressure Buildup—The first method of eliminating pulsation suggested by the mechanism was to make the flame less sensitive to changes in

air delivery. One way of doing this is to eliminate the driving force, that is, to stabilize the flame with a flame holder. This device is an obstruction placed in the combustion air stream which sets up a stagnation zone, acts as a pilot, and thus prevents the flame from "blowing" downstream. With the flame thus anchored, the changes in air delivery caused by pressure surges do not have such a drastic effect on the flame and acoustical energy does not build up in the system. Pulsations are therefore eliminated.

Unfortunately, this solution is not practical in intermittently operated oil flames because of coke buildup when the unvaporized oil droplets strike the cool metal flame holder. Home oil burners operate intermittently, and such deposits would create servicing problems. On the other hand, this solution seems practical in commercial boilers which operate continuously because the coke would be removed as fast as it forms.

Venting Relieves Pressure Buildup and Stops Pulsation – The second method of eliminating pulsation suggested by the mechanism was to dampen the acoustical system and thereby minimize changes in the flow of incoming combustion air. This can be done most effectively by venting the combustion zone to the atmosphere.

Venting was provided on the laboratory furnace by providing twelve ½-in. diam holes into the combustion chamber. To accomplish this, a "venting plate" shown in Fig. 4a was attached to the burner blast tube. The location of the venting plate in relation to the furnace is shown in Fig. 4b. With this plate installed, pulsation was eliminated and noise level was reduced substantially. These results are shown graphically in Fig. 5 where the db measured in each octave band are plotted for both a pulsing and a nonpulsing flame. With pulsation eliminated, the entire sound pressure level gradient is reduced, but even more significant is the absence of the most annoying component in the 20-75

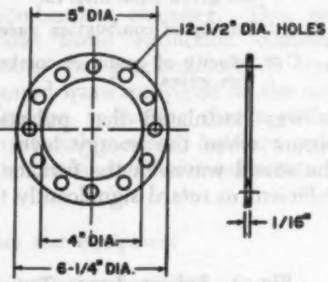
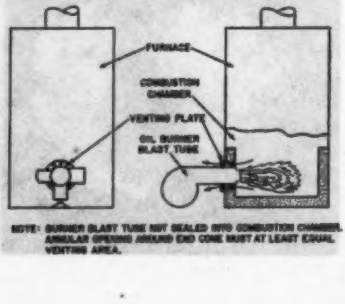


Fig. 4a Sketch of anti-pulsing venting plate

Fig. 4b Anti-pulsing venting plate installed



cps octave band. This component is reduced from 92 to 76 db which cuts the relative loudness in half.

When the venting area was moved to locations further from the combustion zone, the ability to reduce fluctuations in incoming combustion air was less. This was determined by the area required to relieve pulsation at two locations in the furnace. When vented at the combustion chamber only one sq in. of area was necessary to relieve the pulse. However, when vented in the heat exchanger, three sq in. of area was required. Large areas, of course, lower efficiency by admitting excess air. These data are shown in Table I and demonstrate the effectiveness of locating the vent holes at the combustion chamber.

When the combustion chamber is vented, excess combustion air is admitted through the venting holes. The question then arose as to whether the pulsations were eliminated by pressure relief or by the addition of excess air. Addition of excess air through the burner blast tube is known to eliminate pulsation, because of the increased pressure drop available to retard the advance of the oncoming pulsation or standing wave. Therefore, if the mechanism is pressure relief, less excess air should be required to eliminate pulsations.

Experiments showed that when air was supplied through a manifold to these vent holes, the total amount of air required to eliminate pulsation was increased. However, when the vent holes were opened to the atmosphere, the total amount of air required to eliminate pulsation was decreased. These data show clearly that the effect of the vent holes was primarily an acoustic one. Test data are assembled in Table II.

Another possibility is that the venting holes altered the resonant

Fig. 5 Elimination of pulsation reduces burner noise level

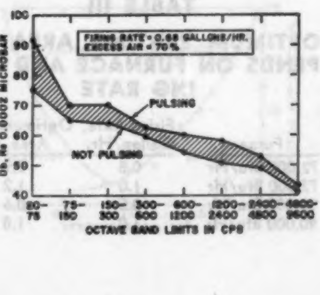


TABLE I

VENTING HOLES MOST EFFECTIVE IN COMBUSTION CHAMBER

Vent Hole Location	Vent Area, In.² Required for No Pulse
Combustion Chamber	0.98
Heat Exchanger	3.14

TABLE II

PRESSURE RELIEF, NOT EXCESS AIR ELIMINATES PULSATION

Air Delivered Thru Manifold, SCFM	% Excess Air Required to Eliminate Pulsation — Vent Open	Vent Closed
0	39	65
1	37	69
2	43	73
3	46	80

frequency of the system and prevented pulsation. However, laboratory tests showed that these holes did not change the pulsing frequency.

It was concluded from these data that venting stops pulsation by pressure relief and not by excess air addition or changes in resonant frequency of the system. This conclusion was confirmed by later experiments, where different diam holes and thickness of pressure relief plates were used to achieve the same venting area. The plates with the smallest pressure drop through the venting holes eliminated pulsation with the least amount of excess air, indicating that more acoustic pressure was relieved. Other acoustic changes to the furnace did not change the pulsing frequency.

Venting Plate Is A Practical Anti-Pulsing Device – A good, practical venting device must provide:

1. Adequate pressure relief from the combustion zone to eliminate pulsation.
2. Minimum losses in efficiency.
3. Ease of installation and adjustment.
4. Low initial cost.
5. Adaptability to most heating units.

The simplest device found in our laboratory studies that meets these requirements is the venting plate.

Twelve ½-in. holes are available for pressure relief, if necessary. Different units and firing rates have different venting area requirements, as shown by the data listed in Table III. These laboratory tests on two warm air furnaces indicated, however, that twelve ½-in. holes are ample for small, pulsation-prone, domestic furnaces. The laboratory units had capacities of 72,000 and 90,000 Btu/hr and were fired over a range of 0.5 to 1.0 gph. In no case were more than nine holes needed to eliminate pulsations. In larger units using higher firing rates more energy release is involved and hence venting area requirements could be greater. However, larger units are less prone to pulsate and further tests are needed to substantiate the extrapolation above one gal/hr.

Loss In Efficiency Is Negligible – Although venting holes prevent pulsation by relieving acoustic pressure, they can admit a small quantity of air to the combustion zone and thereby reduce efficiency. The second requirement of a practical venting device must be to minimize these losses in efficiency.

Laboratory tests were therefore made to determine the maximum efficiency of a test unit using the venting technique. A series of

TABLE III

OPTIMUM VENTING AREA DEPENDS ON FURNACE AND FIRING RATE

Furnace	Firing Rate, Gallon/Hr.	Optimum Vent Area, In.^{2}
72,000 Btu/Hr	0.5	0.9
72,000 Btu/Hr	1.0	1.2
90,000 Btu/Hr	0.5	0.6
90,000 Btu/Hr	1.0	1.8

runs were made at various venting areas ranging from 0 to 2.0 $in.^2$ At each venting area, the primary air shutter was gradually closed until either pulsation or smoke occurred. The excess air was measured at this first incipient point.

At low venting areas, pulsation limited the maximum obtainable efficiency. As the venting area was increased, the pulsing tendency was decreased, which allowed the burner to operate with less excess air and efficiency increased. These data are shown in Fig. 6a. When the venting area was increased still further, too much air was admitted through the vent holes, which caused inefficient air-oil mixing and inefficient combustion. Hence, at large venting areas satisfactory operation of the burner was limited by smoke formation rather than pulsation.

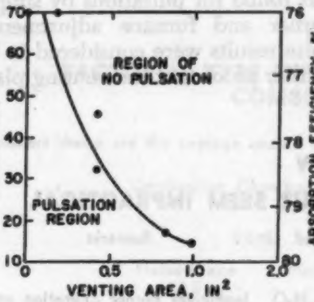
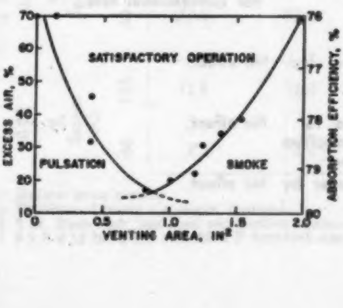


Fig. 6a Increasing venting area lowers excess air required to stop pulsation

Fig. 6b Too much venting area causes inefficient use of air



These data also plot as a smooth curve and are shown in Fig. 6b together with the previous pulsation limiting curve. The intersection of these two curves represents the optimum venting area since it is the maximum efficiency attainable without either smoke or pulsation occurring. It is less than 1% lower than the efficiency obtained with the same burner in a pulsation-free furnace. This small loss in efficiency should not be serious, and from these data it was concluded that the venting plate meets the second requirement of minimizing efficiency losses.

The other design requirements, ease of installation, low cost and adaptability to most heating units, are satisfied by the simplicity of the thin plate. This low cost device is installed merely by sliding it onto the blast tube and sealing it to the front of the furnace. The only time an installation would be more involved would be when the burner is sealed to the inside of the combustion chamber. In such cases, an opening would have to be provided in the combustion chamber around the blast tube equal or greater in cross-sectional area to that required for optimum venting. A diagram of a

typical installation is presented as Fig. 4b.

Other Techniques Can Work But No Practical Ones Were Found – Many attempts were made to eliminate pulsation by simple burner and furnace changes. The techniques used, the range tested and the results are tabulated in Table IV. Although most of the changes were capable of decreasing the tendency to pulsate, none eliminated it entirely without reducing combustion efficiency substantially.

Those changes that did not decrease efficiency seemed impractical for existing burner installations. For example, the tendency to pulsate was decreased by increasing the air pressure drop across the burner end cone. Unfortunately, conventional burner fans do not build up high enough static pressures to eliminate pulsations entirely. In our tests, the maximum obtainable pressure drop of 0.25 in. H_2O was insufficient. Even when 0.50 in. H_2O was developed with the aid of a supplementary blower, pulsation occurred. Sanders and Lawrie[4] recommend a fan capable of developing 2 to 3 in. H_2O to insure pulsation-free operation for oil burner operation.

Because no sure-fire remedy was found for pulsations by simple burner and furnace adjustments, these results were considered to be further proof that the venting plate

TABLE IV
OTHER TECHNIQUES WORK BUT SEEM IMPRACTICAL

Technique	Range Tested	Remarks
1. Decreased Tendency to Pulsate By:		
Increasing Draft	0.015 to 0.130 in. H_2O	Inefficient burner operation and required draft too high.
Increasing Excess Air	0 to 70%	Inefficient combustion at 70%.
Changing Atomizing Air*	4 to 8 in. Hg Pressure	Noisy or inefficient combustion.
Increasing Oil Pressure	60 to 140 psi	Impractical increases required.
Lowering Firing Rate	0.90 to 0.50 gph	Required rate too low for practical use.
Acoustical Damping	Various dashpots and surge chambers	Size of devices impractical in field.
Increasing Air Pressure Drop Through Burner	0.04 to 0.50 in. H_2O	Required pressure drop too high for conventional fans.
2. Pulsation Not Affected By:		
Changing Oil Distribution Pattern of High Pressure Nozzle	Symmetrical to lop-sided	No effect.
Recirculating Combustion Products	Baffles installed to shorten or lengthen residence time	No effect.
Eliminating any Possible Fluctuations in Oil Feed*	Fed oil to burner by gravity	No effect.

* Low pressure, air atomizing burner only

is the best practical anti-pulsing device.

Easily Controlled Flame "Variables" Studied—The second phase of the burner noise study was concerned with non-pulsing flame noise or "combustion roar". As shown previously, the flame itself is the major contributor to high noise levels. The problem of determining how to reduce this noise was approached by detailed consideration of the operating variables subject to change, that would alter flame characteristics. The variables considered were:

- Firing rate
- Flame pattern
- Excess air
- Draft
- Combustion chamber design
- Nozzle design
- Fuel characteristics

The fuel characteristic effect was described in a previous paper[1] and is not discussed herein. Suffice it to say that major changes in fuel chemical composition, and volatil-

TABLE V
FACTORIAL-TYPE EXPERIMENT FOR NONPULSING COMBUSTION NOISE

Numbers shown are the average overall sound pressure levels, in decibels, for several runs.

Combustion Chamber, 9 I.D. x 12 in. High, Hard Pre-Fab Material
Draft, In. H_2O Measured Over Fire

Firing Rate, gph	% Excess Air	0.015 Flame Shape 30 F	0.015 Flame Shape 80 F		0.050 Flame Shape 30 F	0.050 Flame Shape 80 F	
1.00	125	73.8	78.2 69.2[3]	69.8[2]	74.3	76.1	69.0[2]
1.00	40	76.0	79.1		71.9	75.8	
0.75	125	75.5	75.8 73.3[4]	76.5[1]	71.8	72.8	73.8[4]
0.75	40	73.6	77.0	76.7[4]	72.7	76.8 76.5[3]	75.5[1] 77.6[4]
0.50	125	73.5	72.0		71.0	70.3	
0.50	40	X	73.8		X	X	

(1) Hollow cone nozzle.
(2) Position of inner air sleeve changed.
(3) 7 in. diam stainless steel combustion chamber.
(4) 8 x 8 x 12 in. high, soft 2000 F firebrick combustion chamber.

ity, gave no significant change in noise levels.

Since the variables listed above would not necessarily be independent in nature, a factorial-type experiment was designed so that possible interactions between the variables could be evaluated. This experiment is described by Table V, which also includes the data that were obtained. A prime consideration was to determine if some change or a combination of changes could be made by the manufacturer or servicemen so that oil burner noise levels would be lowered.

Lowering Firing Rate Gives Most Noise Reduction—Firing rate had the greatest effect on noise level of all the variables tested. Firing rates of 0.5, 0.75, and 1.00 gph were used since this range brackets the heating requirements of an average house. Noise level decreased consistently with firing rate and typical results are shown in Fig. 7.

The 8 db decrease made possible by reducing firing rate from 1.0 to 0.5 gph (80 F flame) is significant. It is similar to the sensation of leaving a room with a window air conditioner in operation, but still remaining in a close-by room or hallway.

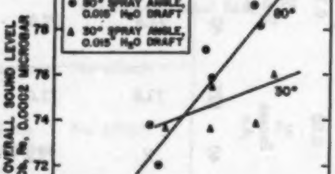
Reducing the firing rates of burners in the field will effectively reduce the noise level and at the same time may give increased efficiency by better utilization of the heat exchanger. The magnitude of this change, however, depends on two factors: the lowest possible firing rate that will supply the house with enough heat, and the firing rate of the burner in question. Noise data were not obtained by reducing firing rates initially above 1.0 gph. While a noise reduction will occur, its magnitude has not been determined.

Narrowing Flame Pattern Also Reduces Noise—Fig. 7 also shows the effect of narrowing the flame pattern from 80 to 30 deg. The narrow 30 deg flame was 1 to 5 db quieter than the wide sunflower 80 deg flame at firing rates of 0.6 and 1.0 gph, respectively. At lower firing rates the effect of flame pattern disappeared, probably because both lower firing rate and narrower oil patterns are decreasing noise by the same route.

The narrowing of flame pattern is a practical means of lowering combustion noise. Although the amount of noise reduction obtainable is a function of minimum spray angle, two practical limitations exist:

Inside geometry of furnace, and

Fig. 7 Non-pulsing noise reduced by lowering firing rate



Burner adaptability to narrower air patterns.

The depth of the furnace must be sufficient to accommodate a long narrow flame and the burner must be capable of providing a narrow air pattern. Close matching of air and oil patterns are necessary for efficient operation. In most cases, air patterns can be matched to narrow oil sprays by using a smaller end cone on the burner. Tests were also made with hollow cone nozzles which gave results similar to the solid type.

Other Changes to Flame Are Insignificant or Impractical Noise Reducers—The variable of excess combustion air was studied at two levels, one representing acceptable operation at 40% and the other representing inefficient operation at 125%. No significant trend was observed. Combustion chambers of different size, shape, and materials of construction were tested also. These consisted of round and square combustion chambers of equal and different cross-sectional area. Stainless steel, soft 2000 F firebrick and a hard prefabricated chamber were tried. All cases had essentially the same noise level of 77 db, indicating that combustion chamber characteristics are not critical.

Two "over the fire" draft conditions that can be obtained easily in the majority of domestic installations were used in the variable study. These were the minimum acceptable standard of 0.015 in. H_2O and a much higher setting of 0.05 in. H_2O. Higher drafts are impractical in most furnaces because of the accompanying loss in absorption efficiency caused by air leakage. Although the results were consistently in favor of the higher draft, the noise level reduction was only 2 db, barely above that detectable by the human ear. Because of this small decrease and the lower efficiency of high draft, the use of higher draft is not a desirable method of reducing noise.

In addition to the variables examined in the factorial-type experiment, the effect of flame holders was examined because of a lead that turned up during the pulsation studies. The high speed motion pictures showed that even nonpulsing flames have oscillating flame fronts. Such oscillation or momentary instability generates sound waves which contribute to overall flame noise. Consequently, a series of tests was run with several different types of flame holders, which reduced overall flame noise from 3 to 6 db. However, as mentioned previously, flame holders seem impractical in intermittently operated oil burners. In view of these results, further development work to make the device practical and prevent coke buildup may be worthwhile.

Findings Can Be Used by Industry — The findings of these studies can be used by equipment manufacturers and servicemen to minimize oil burner flame noise.

By far the biggest noise reduction can be effected by eliminating pulsation. This is accomplished easily and without losing efficiency

by using the venting plate. In regard to its use, our studies covered only a small segment of oil-fired installations.

Our results are related specifically to the units tested, but the principle can be applied to oil-fired units in general. These units fall into two classes: new units where manufacturers can modify the equipment before marketing and field installations where servicemen are called upon to correct pulsation.

Manufacturers should follow the same procedure used in the laboratory and discussed in this paper. In brief, test units with three or four of the most popular makes of burners and determine the venting area required for nonpulsing operation at all operating conditions. Plates can then be designed as part of the unit or as an attachment with sufficient holes to handle all cases.

Servicemen should have two or three plates available: one identical to our test plate and the others with larger venting areas. Simple trial and error in a few installations would supply the necessary experience to minimize future work.

With pulsation eliminated, the noise level is reduced 16 decibels.

In the majority of oil burner installations, however, pulsation does not exist and consequently the noise reduction that the serviceman can accomplish is not as great. By lowering the firing rate and using a narrower spray angle, the flame noise can be reduced 5 to 8 db. Since lower firing rates also provide higher efficiencies, this combination seems desirable. As a result of these changes, the relative loudness of the oil burner was reduced to approximately the same annoyance level as a window air conditioner except that the oil burner is in the basement. Because of the nature of the gun burner in regard to the high air turbulence and unstable flame, any further reduction in noise through simple equipment changes seems unlikely.

ACKNOWLEDGMENTS

The authors express appreciation to W. A. Beach and J. V. Gale, both of the Esso Research and Engineering Company, for their contributions to this work.

REFERENCES

(1) Effect of Fuel Composition on Oil Burner Noise, W. A. Beach, R. W. Sage, H. F. Schroeder, ASHRAE JOURNAL, July 1959.

(2) Noise in the Modern Home, E. E. Mikeska, Noise Control, May 1958.

(3) A Community's Reaction to Noise: Can It Be Forecast?, Stevens, Rosenblith and Bolt, Noise Control, January 1955.

(4) Studies of a Diesel Driven Heat Pump, H. H. Vickers and R. W. Sage, Heating, Piping and Air Conditioning, May 1957.

(5) Low Frequency Combustion Noise in Oil-Burning Equipment, G. J. Sanders and W. C. Lawrie, Heating, Piping and Air Conditioning, October 1958.

(6) Pulsations in Residential Heating Equipment, A. A. Putnam and W. R. Dennis, Heating, Piping and Air Conditioning, July 1956.

DISCUSSION

NORMAN A. BUCKLEY, Cleveland, Ohio (Written): It would appear that it is in the area of flame stabilization that the greatest possibility lies for reduction of oil flame noise level.

The graphic presentation of noise levels, comparing oil burning noises to refrigerators, dish washers and vacuum cleaners is commendable, but it seems to point out a basic weakness in our methods of noise level analysis. However, if this is a scale on which understandable comparisons can be made, we should continue working for comparison to the new refrigerator and not accept the window air conditioner to dish washer range.

Studies should not be restricted to what can be accomplished by a field service man. If increased overfire draft of 0.05 in. of water reduces noise level, what can be accomplished with 0.10 or 0.30 in. of water overfire draft?

Studies reported were restricted to hard surface pre-fab type combustion chambers. Are there any comparable data on what can be accomplished with light refractory or ceramic felt blanket type chambers?

AUTHOR SAGE: Some work was done in the field of stabilization with what we call flame holders, such as a ball on the end of a rod. Flame stabilization seems to hold the flame right back at the tip of the blast tube. However, this probably would be difficult to put into operation in the field without considerable study.

Regarding the comparisons with other air conditioners, we were trying to give some kind of mental comparison with something people might have heard, because it is likely that many people have never been subjected to it consciously.

On the subject of more overfire draft, it does not seem likely that this would be a change possible in the majority of systems in the field. These research studies were concentrated towards trying to do something for units already in the field. Fans that produce 3-4 in. of water are fine but there are not many of them and they are expensive.

Only limited data are available on soft fire brick chambers. With the soft fire brick there were reductions of from one to three decibels over the entire frequency range compared with the hard fire brick. However, this was at the 125% excess air level. When we went down to 40% excess air, which is where a burner should be operated, it was not possible to distinguish between the soft and hard fire bricks. Pulsation is at the low frequency end of the spectrum and this makes it difficult for an acoustical fibre to work effectively.

D. W. LOCKLIN, A. A. PUTNAM AND C. F. SPEICH, Columbus, Ohio (Written): Two effective means for suppression of pulsations have been highlighted by the author: the use of flame holders to stabilize the flame and venting of the combustion chamber to reduce the acoustic pressure. A more scientific basis for the field practice of venting has been provided

An important contribution of this paper is the conclusion that the benefits of venting in the suppression of pulsations lie in the acoustical influence rather than in the effect of bleed air introduced into the combustion process. Because of the significance of this finding, it would be helpful if the authors would amplify their discussion of the technique used in obtaining the data in Table II.

In Table IV, the alteration of the residence time for the recirculated products is shown to have had no effect on the amplitude of pulsation. We would be interested in some additional explanation of how the residence times were changed. Was the combustion-chamber volume or throughput of combustion products altered during the studies, as would be necessary to change the mean residence time of the fuel and air products in the combustion chamber? Perhaps the time referred to is the transit time along a recirculation path. It is interesting to note that several types of burners using controlled recirculation of combustion products are said to operate extremely quietly. Such units usually utilize larger combustion chambers than conventional burners of similar firing rate. Thus, the mean residence times for the combustion products would be longer.

Near the end of the paper, where the findings are interpreted in terms of their usefulness to the industry, an observation is made concerning the amount of noise reduction when pulsation is eliminated. Although the noise reduction is certainly significant in magnitude, it is questionable whether a 16 db reduction will accompany the suppression of pulsation in all cases.

In Fig. 7, the variable of excess air appears to contribute to some scatter of data. If scatter bands are visualized for the two spray angles, then two bands tend to merge at the lower firing rates. The authors may wish to comment on whether this tendency may be indicative of the fact that, in situations where combustion rate per unit volume is reduced, the spray angle has less influence on the combustion noise.

AUTHOR SAGE: Our study was not oriented at low pressure burners. It happened that we became especially interested in pulsation while testing a low pressure burner, and we did use that low pressure burner as one of the units in the experiments performed. However, in general, the work was pitched at either low or high pressure burners and pulsations were obtained in high pressure burners.

In connection with the question of venting, it was firmly established that venting works by acoustical effect rather than by the introduction of additional air.

It seems to me that this db reduction is true. In our furnaces we did get such a reduction. While these results do not necessarily apply to all systems, it is estimated that they do apply to about 95% of the systems in the field today.

In connection with our basic objective, it seemed to be impractical to run an oil burner with less than stoichiometric air.

Charles Schrader, Cleveland, Ohio: In our experiences, pulsation has always been a problem.

Moving up into the commercial and industrial sizes, in keeping with the present-day tendency toward stacks, positive furnace pressures are obtained and, as a result, venting can not be used. Therefore, pulsations must be dealt with by some other means, usually some change in the configuration on the choke range shape.

Shape has been found to be important. Changing spray angles helps but usually if that is done, then something else has to be changed in order to get good combustion. We have had good results by working with high pressure drops across the burner head. This is not a case of just taking a conventional burner and reducing the choke ring size; all parts must be changed.

Our differentials are in the neighborhood of about an inch of water pressure. This has been experienced not only in the small burners of five or six gal but also in those of 100 gal or more.

G. Applegate, Trowbridge, England: Was the combustion chamber changed or modified? Has any research been carried out in the prevention of carbon formation in the boiler and in cutting down stack noise? Have you had any experience with regard to ratio absorption of heat from the flame as direct radiation in relation to pulsation noise?

Author Sage: In all of our studies we felt it was impractical to come up with a solution that would wind up with a smoking burner. Every time a change was made, another change was made to prevent the formation of smoke if it had occurred.

This is important in this study because it means that as we changed the angle of the spray nozzle, we had to change the air pattern. In some cases, when the flame angle was changed, it was necessary to change the combustion chamber. For example, in one test we moved the back wall of the chamber because of flame impingement. It is true that this could have an additional effect by changing the recirculation of gases in the chamber.

With regard to the absorption of heat from the flame. We attempted to correlate the volumetric heat release rate with pulsation and with nonpulsing flame noise. The data were inconclusive.

The draft regulator was located all in the flue at various times in an effort to determine its effect. Various dampers were used in the stack and this was found to have little effect. The trouble was in the furnace.

1749

No. 1749

Influence of the House on Chimney Draft

A. G. WILSON
Member ASHRAE

Draft is the pressure difference between some point in a venting system and the surrounding air at the same level. It is common to consider draft with respect to outside air, when predicting the draft provided by residential chimneys. But the draft between the base of the chimney or firepot and the surrounding inside air is the one effective in venting connected appliances. Draft requirements and chimney design are usually based on conditions expected when connected appliances operate steadily at rated output, when the difference between draft with respect to outside and inside air may be unimportant. However, a number of appliances, for example, solid-fuel hand-fired furnaces or oil units with pot-type burners, operate at low fire much of the time. Even with gas or mechanically fired oil-burning units there is a period of non-steady flue gas temperature at the beginning of each on-cycle.

Some of the venting problems that arise under these conditions can be understood better by considering the relation between chimney draft and house pressures. In this paper, this relationship is examined and demonstrated by results of some field measurements. Its application to a specific case of venting failure with solid-fuel hand-fired furnaces is discussed. Included, too, are results of field measurements of draft during start-up of a furnace with a high-pressure gun-type burner.

VENTING FAILURE AT LOW FIRING RATE

In the spring and fall, when houses have low heat requirements, draft problems with heating units which can operate on low fire are not uncommon. Appliances under hand control such as oil burning space heaters or solid-fuel hand-fired furnaces, where the combustion rate

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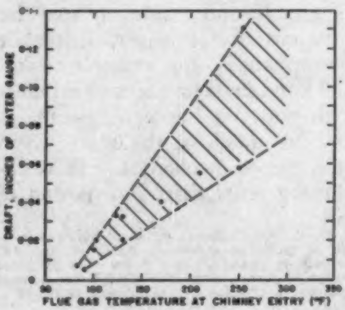
is modulated roughly to conform to heat requirements, are in this category. Sooting of flue passages with the former and venting failure and fume poisoning with the latter are sometimes reported. These venting failures are often ascribed to down-drafts or unusual atmospheric conditions.

However, in recent investigations in two widely separated housing developments, venting failures were reported during calm, mild periods, often during sleeping hours. The houses, 1½-story units with basements, had gravity warm air heating systems and hand-fired coal-burning furnaces venting into standard lined masonry chimneys located on an exterior wall. Chimneys were 25 ft high with 7⅛-in. square flues inside; smokepipes, about 12 ft long contained a plate damper. Pennsylvania anthracite coal was burned. Venting failure was not reported in similar bungalow units with chimneys located inside the house.

In mild weather ash pit dampers were generally left closed, and air for combustion was admitted through a manual slide in the firing door. Combustion was controlled by manipulating the smokepipe plate damper. Measurements of flue gas temperatures and chimney draft were taken in a number of houses with furnaces operating at low combustion rates. With the smokepipe damper closed, temperature drops of 40, 90 and 140 F occurred in the smokepipes with gas temperatures at the flue collar of 150, 250, and 350 F respectively. With the smokepipe damper open, temperature drops with the same gas temperatures at the flue collar were increased to 60, 150, and 220 F due to diluent air.

Fig. 1 gives results of the draft measurements related to flue gas temperatures at chimney entry. Values fall within the cross-hatched area; some spread is expected as measurements were taken under differing conditions of flue gas volume. Outside temperatures during the measurements were between 25 and 34 F. Basement temperatures were near 70 F. Draft at the smokepipe plate damper was not measured but would have been less than at the chimney by the amount required to overcome smokepipe pressure losses. Thus, under the conditions shown at the lower end of the graph, venting failure was imminent. Some of the factors contributing to these conditions are apparent, but the actual mechanism by which venting failure occurred can be fully explained

Fig. 1 Draft vs. flue gas temperature at chimney entry



only by considering the inter-relation of draft and house pressures.

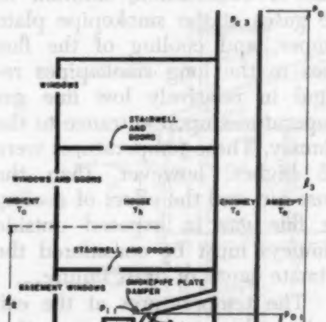
INTER-RELATION OF DRAFT AND HOUSE PRESSURES

Fig. 2 is a simplified diagram of a two-level house with basement and attached chimney. Elevation l_2 represents the level of the neutral zone where the absolute pressures inside and outside are equal. Elevation l_1 represents the location of any opening between the venting system and the house, such as a smokepipe plate damper, a barometric damper, or a draft hood. Venting failure occurs when the pressure in the venting system at this location, p_{e1}, is greater than the pressure in the house at the same level, p_{i1}. Since pressure p_{e3} at the chimney exit is equal to the outside pressure p_{o3} at the same elevation, l_3,

$$p_{e1} = p_{o3} + w_e (l_3 - l_1) + p_{fe} \quad (1)$$

where

Fig. 2 Simplified diagram of 2-story house



w_e = mean weight of flue gas, lb per cu ft

p_{fe} = pressure loss in the chimney due to flow resistance, lb per sq ft

The outside pressure at elevation l_1 is

$$p_{o1} = p_{o3} + w_o (l_3 - l_1) \quad (2)$$

where

w_o = mean weight of outside air, lb per cu ft

The chimney draft with respect to outside air at elevation l_1 is then

$$p_{o1} - p_{e1} = (w_o - w_e)(l_3 - l_1) - p_{fe} \quad (3)$$

Similarly, the chimney draft with respect to inside air at elevation l_1, is

$$p_{i1} - p_{e1} = w_o (l_3 - l_2) + w_i (l_2 - l_1) - w_e (l_3 - l_1) + p_{fi} - p_{fe} \quad (4)$$

where

w_i = mean weight of inside air, lb per cu ft

p_{fi} = pressure loss in the house due to resistance to flow between elevations l_1 and l_2

The resistance to flow between levels within the house is usually negligible and the difference between the chimney draft with respect to outside and inside air is

$$p_{o1} - p_{i1} = (w_o - w_i)(l_2 - l_1) \quad (5)$$

Equation (4) shows that venting failure and spillage of flue gas into the house from openings at elevation l_1 will occur if

$$w_o (l_3 - l_2) + w_i (l_2 - l_1) \leqq w_e (l_3 - l_1) + (p_{fe} - p_{fi}) \quad (6)$$

Thus the location of elevation l_2, the neutral zone, has an effect on draft with respect to inside air and is related to draft failure. If elevation $l_2 = l_1$, the draft with respect to inside and outside air will be

equal at elevation l_1 and will be zero when the mean temperature of the flue gas in the chimney, T_c, is equal to the outside temperature, T_o, ignoring friction effects.

Similarly, if elevation $l_2 = l_3$, the draft will be zero when T_c is equal to the mean temperature in the house, T_i. If there is flue gas flow in the venting system, friction losses must be overcome. These limits of mean flue gas temperature, below which venting failure occurs, will be correspondingly higher. An increase in the mean flue gas temperature of approximately 10 F would normally be adequate to overcome these friction losses with the small flue gas volumes involved when venting failure is imminent.

Wind pressures may affect the neutral zone location, and, in addition, will usually have some direct effect on pressures at the chimney exit. Wind effects are beyond the scope of this paper. Under calm or low wind conditions, the level of the neutral zone will usually be at some elevation between l_1 and l_2 and will depend on the location and characteristics of the openings between the house and outside.

Data on the location of the neutral zone in houses are limited. Available records indicate that it may often be well above the mid-height of the heated structure. It follows that opening basement windows will have a beneficial effect on draft, but openings at upper levels may affect draft adversely. Appliances that exhaust air from the house have the same effect as openings above the neutral zone. In a tight house these might lower house pressures below outside pressures at all levels, creating an imaginary neutral zone above the heated structure. It will be recognized that the chimney itself represents one of the upper openings. In a tight house or enclosure the chimney might have an effect on the neutral zone similar to a mechanical exhaust unit.

The mechanism of draft failure described previously applies directly to conditions found at the housing developments referred to. It can be assumed that the neutral zone was at an upper level in the house, particularly at night when upstairs windows are open, and at times the mean flue gas temperature fell below the critical value defined by Equation (6). It is clear that to avoid this cause of venting failure the mean flue gas temperatures in the chimney cannot fall much below mean house temperature.

Several factors contributed to the lowering of mean flue gas temperature below this value. Low rates of combustion, dilution of flue gases at the smokepipe plate damper, and cooling of the flue gases in the long smokepipes resulted in relatively low flue gas temperatures upon entrance to the chimney. These temperatures were still higher, however, than the house air, and the effect of cooling the flue gas in exposed outside chimneys must be considered the ultimate cause of draft failure.

The temperatures at the exit of the chimneys where venting failure occurred were not meas-

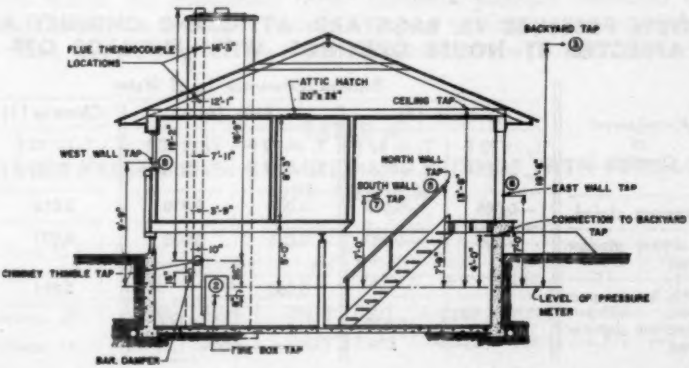
ured. Subsequently, however, some temperature records were obtained for three chimneys of similar construction venting oil burning furnaces. Two of these chimneys were on outside walls, one 27 ft and the other 19 ft high. The third was an inside chimney 22 ft high; 12 ft within the heated structure, 7 ft in the attic, and 3 ft above the roof. Following an 8-hr burner-off period, when outside temperature was 40 F, the flue gas temperatures at chimney exit were 47, 48, and 65 F, respectively. Venting failure of the type described, with solid-fuel hand-fired furnaces, is unlikely to develop when inside chimneys are used.

Similar venting problems can be expected when heating units burning other fuels operate at low or pilot fire during mild weather, although spillage of the products of combustion into the house might not be regarded so seriously. It follows that inside chimneys would also be advantageous in these instances. On the other hand, mechanical oil-burning furnaces operate at rated input on heat demand, regardless of the outside temperature, and the same problem of venting failure due to low flue gas temperatures does not occur. However, the draft condition during start-up of such furnaces, and the extent to which this is affected by the draft prior to start-up, are of special interest. Since the average temperature of gas in the chimney can be relatively low, the effect of house pressures on the chimney draft prior to starting of the burner is next considered.

MEASURING CHIMNEY DRAFT AND HOUSE PRESSURES

Measurements of chimney draft and house pressures were made in a single-story house with basement, heated by a forced warm-air system with high-pressure gun-type oil burner. The house had a floor

Fig. 3 Location of pressure taps and chimney thermocouples



area of 1200 sq ft and was well insulated, with plaster finish throughout. Storm windows and doors and weatherstripping were used. There was a covered hatch in the hall ceiling leading to a vented attic.

The chimney was on an outside wall and exposed to the weather on three sides. It was constructed of single layer brick with clay-tile flue liners and contained a nominal 8- by 8-in. flue serving the furnace and a nominal 8- by 8-in. flue serving a fireplace. The furnace room was open to the basement which was interconnected to the main floor through the return air system. The oil burner was equipped with a 0.75-gph nozzle and the air supply was adjusted to give No. 2 smoke spot with about 5% CO_2. The flue pipe was 6 ft long and contained a barometric damper, which was partly open even when the furnace was off.

Pressure taps were located in the cap of the combustion chamber sight-port to measure overfire draft, in the flue pipe at chimney entry to measure chimney draft, in the four outside walls and ceiling to measure pressure differences across the building enclosure, and at an outside pressure station in the backyard approximately 80 ft from the rear of the house. Pressure measurements were made with an electric capacity-type differential pressure meter. This provided a current output proportional to pressure, which was measured on a galvanometer-type recording milliammeter. The pressure meter was sensitive to pressure charges of less than 0.001 in. of water and was calibrated against a laboratory micromanometer having a corresponding accuracy. The zero-position of the meter, however, drifted with temperature and line voltage fluctuations and frequent checking was necessary.

Flue gas temperatures in the chimney, along the center of the

TABLE I

HOUSE PRESSURE VS. BACKYARD, ATTIC AND CHIMNEY AS AFFECTED BY HOUSE OPENINGS, WITH FURNACE OFF

Arrangement of Openings	Pressure Differences, In. of Water				
	Backyard (3)		Attic (4)		Chimney (1)
	$T_i = 72$ F, $T_o = 27$ F	$T_i = 70$ F, $T_o = 6$ F	$T_i = 72$ F, $T_o = 27$ F	$T_i = 70$ F, $T_o = 6$ F	$T_i = 72$ F, $T_o = 27$ F
Openings closed	−0.005	−0.018	0.005	0.010	0.016
Basement window open	0.000	−0.003	0.016	0.025	0.027
Attic hatch open			0.000		0.011
Fireplace damper open		−0.023		0.006	

flue, and outside temperature were measured with copper-constantan thermocouples connected to a 16-point electronic temperature recorder. Location of the chimney thermocouples with respect to the center of the chimney thimble and location of pressure taps with respect to the level of the pressure meter are shown in Fig. 3. Inside air temperature was measured with a calibrated thermograph at one location only, in the living room approximately 2½ ft from the floor. This has been taken as the mean house temperature in subsequent calculations.

Table I shows, for two different outside temperature conditions, the effects of openings to the outside at different levels on house pressure (at the meter) relative to backyard, attic and chimney pressures. These records were obtained on relatively calm days, during periods when the furnace was off.

When there is no wind, the pressure reading obtained with the meter connected to tap 3 is equivalent to the pressure difference across the building enclosure at the level where the connection from inside to outside is made. If there is wind, the reading merely represents basement pressure with reference to the backyard tap. The reading obtained with the meter connected to tap 4 is equivalent to the pressure difference across the ceiling. The house chimney effect required to produce these pressure differences can readily be calculated from the difference in density between inside and outside air, and a neutral level established with reference to the pressure taps; although where the pressure differences are extremely low, small errors in pressure or air density can lead to significant errors in the calculation.

From the results for an outside air temperature of 27 F, the neutral level, calculated with reference to the attic pressure, is approximately 4 ft below the ceiling with the basement window shut, and approximately 12 ft below the ceiling with it open. This shift of the neutral level below the basement window can be explained by outside wind pressure. There is a corresponding increase in chimney draft with the basement window open. It is anomalous that the basement pressure measured with respect to the backyard tap did not increase to the same extent when

TABLE II

HOUSE PRESSURE VS. CHIMNEY AND OUTSIDE WITH FURNACE OFF AND ON

$T_i = 70$ F $T_o = 6$ F	Pressure Differences, in. of water							
	Chimney (1)	Overfire (2)	Bactvard (3)	Attic (4)	North (5)	East (6)	South (7)	West (8)
Furnace off	0.020	0.019	—0.018	0.011	—0.007	—0.010	—0.008	0.000
Furnace on	0.065	0.035	—0.023	0.007	—0.012	—0.013	—0.012	—0.004

the basement window was opened.

Opening the attic hatch raised the neutral zone to the ceiling level and had a corresponding effect in lowering chimney draft. A similar study of the results for an outside air temperature of 6 F indicates that opening the basement window lowered the neutral level approximately 7½ ft, while opening the fireplace damper raised it approximately 2½ ft.

Table II gives some additional pressure measurements, taken under the same conditions as those in Table I for an outside temperature of 6 F. Calculated neutral levels vary from 3½ ft below the ceiling with reference to the west wall to 1 ft below the ceiling with respect to the east wall, indicating some easterly wind effect. Similarly, the neutral level is 5½ ft below the ceiling with reference to the attic tap and at the ceiling with respect to the backyard tap. With the furnace on and other conditions the same, the neutral level is approximately 2 ft higher with reference to all taps, a result of the increased flow up the chimney.

It should be pointed out in connection with calculated neutral pressure levels, that the house had a total door and window crackage of about 260 lineal ft. Basement windows, not well weatherstripped, represented 35 ft. Living room windows were fixed and ventilation was provided by louvred openings with weatherstripped covers just above floor level, contributing 40 ft. Thus, there may have been more cracks immediately above and below the floor than is usual, and neutral levels may have been correspondingly lower.

Table III presents some further records of the effect of house openings on chimney draft under mild weather conditions. With the furnace off, opening of the basement window raised the draft about 0.006 in. of water, while opening the attic hatch lowered it by about 0.005 in. of water. With the furnace on, the effect of opening the window is somewhat greater, while opening the attic hatch has a lesser effect.

DRAFT DURING BURNER START-UP

Several tests were made with this same installation, in which the chimney or overfire draft was recorded during burner start-up. Fig. 4 gives the results following extended furnace-off periods at different outside temperatures. The accuracy of these measurements of transient draft has not been established. The galvanometer pen was delicately balanced to minimize drag, and the galvanometer was critically damped to eliminate over-

TABLE III

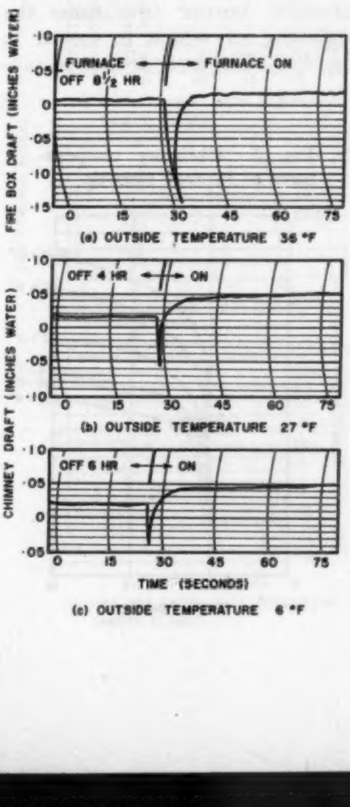
EFFECT OF HOUSE OPENINGS ON CHIMNEY DRAFT

$T_i = 72$ F $T_o = 36$ F	Chimney draft, in. of water	
	Furnace off	Furnace on
Openings closed	0.010	0.066
Basement window open	0.016	0.075
Fireplace damper open	0.005	0.063

shooting. Any errors are thought to be mainly in the measurement of extremes in pressure when pressures were rapidly reversed.

Fig. 4(a) indicates that substantial positive pressures can occur for a brief period in the fire box when the burner-blower first starts. Maximum pressure measured in this test was just under 0.15 in. of water. Draft failure in the fire box occurred for about 5 sec. In this instance, the furnace had been off overnight and the temperature in the center of the flue varied from 69 F at the bottom

Fig. 4 Draft during furnace start-up, following extended off periods



to 49 F at the top prior to burner start-up. The chimney draft was 0.010 in. of water, as shown in Table III. Fire box draft after burner start-up surpassed that prior to start-up in about 6½ sec.

Figs. 4(b) and 4(c) show the extent of positive pressure at the base of the chimney during burner start-up following extended periods with the furnace off. Temperatures from bottom to top of the chimney were 77 to 61 F and 70 to 43 F, respectively. Draft failure at the base of the chimney existed for about one sec in both instances. Chimney draft with the burner operating exceeded that with the burner off after 2½ sec. Draft build-up was relatively rapid in all cases.

There was a corresponding rate of increase in chimney flue gas temperature, as shown in Fig. 5 for the conditions of Fig. 4(c). The curves from top to bottom represent the temperatures at the thermocouple positions shown in Fig. 2, in order, from the bottom to top of the chimney. The print and chart speed of the temperature recorder were not sufficiently fast to show the precise pattern of temperature change during the first few seconds of furnace operation.

As mentioned earlier, the barometric damper was partly open even when the furnace was off. This no doubt led to higher chimney temperatures than would have occurred otherwise, during extended periods with the furnace off. The position of the barometric damper also affected the maximum pressures at the base of the chim-

ney and in the fire box during burner start-up. This can be seen in Fig. 6, which gives the results of chimney and fire box draft measurements with the barometric damper in its normal free position and taped closed. All results in this figure were obtained within a short period under essentially the same conditions.

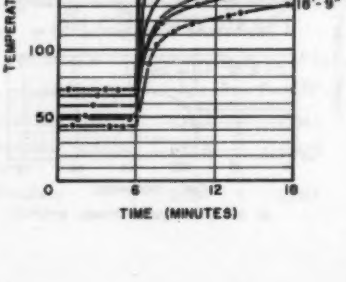
In each case the furnace was allowed to run for less than one min and the system allowed to cool until the chimney draft returned to its original value, with the barometric damper free, before beginning another on-cycle. The outside temperature was 5 F and the burner had been cycling normally before the measurements were taken, accounting for the relatively high chimney draft prior to each burner start.

Higher positive pressure occurred both in the fire box and at the base of the chimney when the barometric damper was closed, indicating that these pressures are relieved, to some extent, by gas flow out of the barometric damper. The rate of draft recovery was somewhat greater with the barometric damper closed, probably due to the higher flue gas temperatures in the chimney, as well as the greater difference between maximum pressure in the chimney or fire box and potential for draft. The position of the barometric damper in this instance had little effect on the period of draft failure, being less than 1 sec at the base of the chimney and 2 or 3 sec in the fire box.

In comparing the results in Fig. 4 with those in Fig. 6, it must be noted that the burner nozzle size was 0.75 gph for the former, but was changed to 0.65 gph for the latter, with a corresponding throttling of the air supply to the burner blower. Thus, both the higher draft before start-up and the reduced capacity of the burner blower probably account for the lower positive pressures and shorter periods of draft failure shown in Fig. 6.

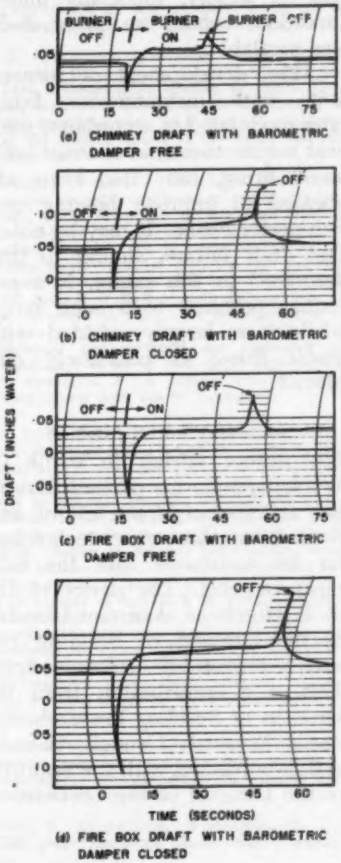
Sudden increase in draft at the moment the oil burner stops, in Fig. 6, is of academic interest. The rate at which the draft subsequently decreases is more important, since it relates to the draft available at the beginning of the next on-period. Fig. 7 is a record of chimney draft following extended burner operation, the beginning of which is shown in Fig. 4(b). The draft decreased rap-

Fig. 5 Chimney temperatures at furnace start-up



idly for the first few minutes but was still nearly 0.03 in. of water after 50 min. Previously, the draft had decreased to 0.016 in. at the end of a 4-hr off-period. In a comparable test with an outside temperature of 6 F, similar results were obtained. The draft fell from 0.065 in. to 0.04 in. in about 15 min but was still above 0.03 in. one hr after the burner stopped. Fig. 4(c) shows that the draft was 0.02 in. of water after a 6-hr burner-off period at this outside temperature. The rate of draft decrease will depend on several variables; these results are, therefore, specific to the installation tested.

Fig. 6 Draft during burner start-up following normal furnace operation with outside temperature at 5 F



CONCLUSION

An analysis has been made of draft failure with residential heating units operating at low rates of combustion during mild, calm weather. It has been shown that under these conditions the relation between chimney draft and house pressures becomes important in determining when venting failure will occur. A simple equation expressing this relation has been developed. It shows that draft failure will occur if the mean flue gas temperature in the chimney falls below a value that depends on the neutral zone level.

Under the most unfavorable conditions, excluding the effect of

Fig. 7 Chimney draft after burner shut-down with outside temperature 27 F

wind, draft failure occurs when the mean flue gas temperature in the chimney is less than the mean house temperature. Measurements have shown that this can occur with masonry chimneys located at an exterior wall and exposed to the weather on three sides, but it is unlikely when chimneys of similar construction have a substantial proportion of their length located within the heated structure.

Measurements of chimney draft and house pressures in a single-story house with an oil-fired furnace have demonstrated that draft, either with the burner on or off, is increased by lowering the neutral zone and decreased by raising it by an amount which corresponds to the change in house pressure at the furnace. The effect of furnace operation on the level of the neutral zone has also been shown. Measurements on a furnace with a high pressure oil burner have established that substantial increases in pressure occur in the fire box during start-up and that positive pressures may exist for several seconds.

Similar increases in pressure, of lesser magnitude, occur at the base of the chimney. The amount of positive pressure and the period over which draft failure occurs depends, to some extent, on the draft available prior to start-up. It appears, also, to depend on the adjustment of the burner blower. Adjustment of the barometric damper affects the positive pressures developed, since it acts as a relief opening. Draft recovery after burner start-up was quite rapid in the installation tested, even following long burner-off periods. Development of draft, following initial draft loss on start-up, was perhaps aided by the burner blower.

It would be interesting to measure the development of draft on start-up with units relying entirely on natural draft for combustion air supply, especially under conditions where no prior draft was available.

The development of furnace units with instantaneous firing rates modulated in accordance with heat requirements, in contrast with on-off firing, has often been advocated to improve heating system performance. It may be noted that draft failure, similar to that described in this paper, is a potential problem with such units, while it is largely avoided with on-off firing as commonly employed.

ACKNOWLEDGMENTS

The author wishes to thank C. Wachmann for his participation in the analysis of draft failure and R. G. Evans, laboratory technician, for his assistance with the field measurements. The advice of Dr. N. B. Hutcheon, Assistant Director of the Division of Building Research, is gratefully acknowledged. This is a contribution from the Division of Building Research, National Research Council, Canada, and is published with the approval of the Director of the Division.

DISCUSSION

D. W. LOCKLIN, Columbus, Ohio: This paper emphasizes the fact that the house must be considered part of the entire combustion system. To improve this situation, there is likely to be a trend toward completely sealed combustion systems to separate the air intake and the exhaust products from the various phases. This trend has been developing in mobile homes and there are reasons for doing this also in residences, especially with the trend toward extremely tight construction.

AUTHOR WILSON: The house is a part of the overall venting and combustion system with most of our present-day heating systems. One of the ways to avoid some of the problems pointed out in this paper is to go to an isolated venting and combustion system.

G. APPLEGATE, Trowbridge, England: In my country, the extremely foggy and humid days have a considerable bearing on drafts. Have any tests been carried out in this country in this direction? Also, what was the orientation of the specific house, in the event that any wind tests were carried out?

AUTHOR WILSON: Tests have never been carried out with conditions of fog on the outside. These conditions do not arise often in our part of the country. With regard to the effect of wind, this is certainly one of the other major variables and one that deserves much more attention than it has received in the past. It is an extremely difficult type of investigation to make. We are attempting to get some useful information from field measurements and currently are examining an installation in which one of the objectives is to determine if the various ways in which wind affects draft can be segregated.

R. B. ENGDAHL, Columbus, Ohio (Written): An additional factor which should be considered in anthracite-fired systems is the effect of flue gas composition on gas density. Usually, it can be assumed that the chimney gas has a density equal to that of air. However, if an anthracite combustion system is tight, so that little dilution of the flue gases occurs, these may become appreciably more dense than air. Theoretically, a sealed system could give a CO_2 content of over 20% by volume. Having a molecular weight of 44, CO_2 weighs approximately 40% more than air. In almost all practical situations, this effect is negligible. But in the case of tight combustion systems burning carbonaceous fuels, especially anthracite or coke, it can be the deciding factor in chimney failure.

AUTHOR WILSON: I agree that in some instances refinements in the draft equations, such as the one suggested, are justified and should be incorporated.

PAUL R. ACHENBACH, Washington, D. C.: This paper illustrates clearly that a heated house acts like a chimney. For a number of years, the GUIDE has contained a statement that parallel flues in the same chimney should not be connected at the bottom because a U-shaped arrangement permits instability. This is in effect what exists here—two chimneys in parallel—one being the house and the other the real chimney with connections near the bottom. If the house itself has large openings near the top, it is more like a chimney.

As pointed out in the paper, the neutral zone can be lowered by increasing the openings near the bottom of the house and this suggests, at least as far as air circulation is concerned, that openings in the house should predominate in the lower floors.

In the case of wind, this same relationship does not necessarily hold true. However, it appears that the openings should still predominate in the lower part of the house but they should be exposed either to prevailing wind direction (the larger openings) or there should be openings to all exposures, so that the pressure in the lower part of the house would tend to equate with the outdoor pressure.

Another possibility is to provide a horizontal barrier at the first floor level so that the whole house can not act as a chimney. However, this is a step forward and an interesting description of how the house can act as a better chimney than the chimney does itself.

W. G. COLBORNE, Windsor, Canada: The kinetic energy or velocity head term should appear in Equation 1. Normally, this term cancels out because the change in velocity head is negligible compared to other terms of the equation. In this paper consideration is given to changes in the neutral zone which are measured in thousandths of an inch of water and also the draft conditions at or near venting failure. Should not the velocity head term be considered and discussed before it is neglected from Equation 1?

The cases in which the velocity head changes might be significant would be in an installation where the area of the flue pipe differed considerably from the area of the chimney; when considerable air is allowed into the flue pipe through the plate damper or through a draft regulator or draft hood; and where over-fire draft became a reference. In this case, the over-fire area is usually large and velocity therefore negligible while the chimney velocity may still be appreciable.

AUTHOR WILSON: A complete equation describing the inter-relationship of house and chimney draft should include a velocity head term. This paper is an introductory one and was intended to demonstrate the inter-relation of house and venting system as simply as possible. It is uncertain whether or not the velocity head term will be significant, but it should be considered.

1750

No. 1750

ASHRAE 68th Annual Meeting, 1961

DENVER, COLO.

The program for the 68th Annual Meeting, held at the Denver Hilton Hotel, June 26-28, included four Technical Sessions, four Symposiums—Industrial Ventilation, Air Conditioning, Food Refrigeration and Domestic Refrigerator Engineering—and seven Forums. There were 15 papers presented at the Technical Sessions; the full texts of these papers, together with discussions upon them, are published in this volume.

Total registration for the Denver Meeting exceeded 800 members and guests.

The Meeting was formally opened at the General Assembly and Business Session on Monday, June 26, at 9:00 a.m. by President R. H. Tull, who expressed his appreciation to the Meeting Arrangements Committee and the Program Committee for their work. R. J. Walker, Co-Chairman of the Meeting Arrangements Committee, speaking on behalf of Fred Janssen, Director of Region IX, extended a welcome to all in attendance. Executive Secretary R. C. Cross announced proposed amendments to the By-laws; these will be voted upon by the membership at the forthcoming Semiannual Meeting in St. Louis, January 29-February 1, 1962.

Perhaps the most significant development at this Meeting was the overwhelming support given by members to a resolution introduced by Research and Technical Committee Chairman E. P. Palmatier supporting the earlier Board of Directors' decision to close the ASHRAE Research Laboratory in Cleveland. Mr. Palmatier proposed the following resolution:

WHEREAS, the Board of Directors, in accordance with its duty under the By-laws to manage the affairs of the Society, has taken action to close the laboratory in Cleveland without a referendum of the membership; and

WHEREAS, subsequent objection to this action by some members has created confusion which is handicapping the effective operation of the Research and Technical Committee in advancing the research program; and

WHEREAS, we have confidence that the Board of Directors has acted and will continue to act in the best interests of the Society in a vigorous research program;

NOW, THEREFORE, I move that the Society support the Board of Directors in these decisions.

Following a discussion from the floor, the measure was put to a member vote. The motion was carried by floor and member vote 4694 for, 31 against; three invalid votes.

Before adjourning this session, President Tull reminded that polls were open for the election of officers and members of the Board of Directors and for voting on

amendments to the By-laws (proposed at the Chicago Semiannual Meeting).

The Welcome Luncheon took place at 12:30 p.m. on Monday, with President Tull presiding. Following the National Anthem, the invocation was delivered by the Rev. W. M. Spence, Pastor of the Lakewood Presbyterian Church. Serving as Master of Ceremonies, L. R. Bindner, President of the Rocky Mountain Chapter, introduced Mr. Janssen and General Chairman V. J. Johnson; both extended greetings and welcome. Mr. Bindner also introduced the Head Table, notable guests, Past Presidents of the Society and Chapter officers.

R. A. Line, Chairman of the Program Committee, presented the speakers at Technical Sessions and Symposiums and Forum Moderators, who rose and were acknowledged. Principal speaker at the Luncheon was Robert T. Person, President of the Public Service Company of Colorado, whose topic was Business Statesmanship.

Reporting on the status of the Society, President Tull outlined briefly the various areas of progress during the preceding months.

Several awards were presented by Mr. Tull. These included:

ASHRAE Klixon Award to George V. Downing, Jr. for his paper, "Thermoelectric Materials", published in the November 1960 issue of the ASHRAE JOURNAL. Sponsored by the Spencer Thermostat Division of Metals and Controls Corp. for the best paper published in the JOURNAL, March through February, on a subject relating to electric motors or controls, the award consists of $150 and a certificate.

Wolverine-ASHRAE Diamond Key Award to Bruno P. Morabito for his paper, "How Higher Cooling Coil Differentials Effect System Economies", published in the August 1960 issue of the JOURNAL. Sponsored by the Wolverine Tube Division of Calumet and Hecla, Inc., a gold key with a diamond inset is awarded to the author of the best paper published in the JOURNAL.

ASHRAE Willis H. Carrier Award to Carl F. Speich, co-author of "Acoustical Systems Determine Oil Burner Pulsations and Their Amplitudes", presented at the Chicago Semiannual Meeting and published in the November 1960 issue of the JOURNAL. This award is sponsored by Carrier Corporation for the best paper presented at a National Meeting by an Associate Member under 30 years of age and consists of $250 plus a scroll.

Mr. Bindner invited everyone to attend the various social functions and urged immediate purchase of tickets. President Tull indicated that election polls closed at 2:00 p.m. and noted that numerous committee meetings were scheduled for that afternoon.

The Banquet was held on Tuesday evening, beginning at 7:30 p.m. President Tull assured the Rocky Mountain Chapter and Region IX of his appreciation of their hospitality. He requested that all Past Presidents and their wives stand and be recognized.

Various honors and awards in recognition of distinguished service to ASHRAE were bestowed by President Tull. Certificates of Appreciation to retiring Chairmen of committees were given to: F. Y. Carter, Advertising Committee; A. Giannini, Research and Fund Raising Committee; F. H. Buzzard, Admissions and Advancement Committee; F. H. Faust, Charter and By-laws Committee; P. B. Christensen, Guide and Data Book Committee; W. L. Holladay, Honors and Awards Committee; R. C. Jordan, International Relations Committee; P. J. Marschall, Meeting Arrangements Committee; K. M. Newcum, Membership Development Committee; C. M. Ashley, Nominating Committee; C. M. Ashley, Professional Development Committee; R. A. Line, Program Committee; G. R. Munger, Publications Committee; E. P. Palmatier, Research and Technical Committee; and P. W. Wyckoff, Standards Committee.

Walter Heywood and Donald Angus, retiring members of the Board of Directors, also received Certificates of Appreciation from President Tull.

Fellow Certificates were presented to: Warren S. Harris, Charles O. Mackey, Alfred J. Offner, George V. Parmelee, Sewell H. Downs, John E. Haines and George L. Tuve.

The ASHAE Homer Addams Award for 1960 was presented to Alan B. Wagner of the Case Institute of Technology. This award is sponsored by the Addams Memorial Fund to be given to a graduate student on an ASHRAE research project. The Committee selects the school and the school recommends the student. The award consists of $600 and a certificate.

ASHRAE's highest honor, the F. Paul Anderson Award for outstanding work or service in heating, ventilating, air conditioning or refrigeration was given to J. Donald Kroeker. Sponsored by the Society, this award is in the form of a gold medal.

Master of Ceremonies John Reed introduced Past President John E. Haines who formally installed the following: Regional Directors—Burt Lomax, Jr., Region VII; P. K. Barker, Region I; J. H. Ross, Region II; E. K. Wagner, Region III. Directors-at-Large—P. R. Achenbach, W. B. Morrison, N. B. Hutcheon and P. W. Wyckoff. Treasurer—John E. Dube. Second Vice President—Frank H. Faust. First Vice President—John H. Fox. President—John Everetts, Jr.

Following his induction, President Everetts presented a Past President's Pin and a framed certificate to outgoing-President Tull. He also gave him a gift, as well as one to Mrs. Tull.

President Everetts outlined the program for the Society during his term. A Dance followed the Banquet.

As is customary at National Meetings, many committees held scheduled meetings during the course of the gathering.

There were various social and special events, also. Among these were technical tours of the Cryogenics Laboratory in Boulder and the United Airlines Flight Training Center in Denver.

The First Technical Session was convened at 9:30 a.m. on Monday, June 26. Professor W. E. Fontaine was Chairman at this session where four papers were presented.

The Industrial Ventilation Symposium was held at the same time. With Robert Jorgensen as Chairman, four speakers participated in this meeting on Laboratory Hood Design. In accordance with established practice, Symposium papers are not published in TRANSACTIONS. Bulletins presently are not planned for any Symposium papers. A condensed version of this Symposium appears in the September issue of the JOURNAL, pages 47-50.

Also concurrent was the Air Conditioning Symposium which considered the topic, Window Units or Central Systems? W. R. Moll was Chairman and there were two speakers.

There was a total of seven Forums. Three were held on Monday afternoon starting at 2:30. The remaining ones took place on Tuesday—one at 9:30 a.m. and three at 2:00 p.m. R. S. Buchanan was Chairman.

The Food Refrigeration Symposium was called to order at 9:00 a.m. on Tuesday. A. L. Brody was Chairman and the subject was Use of Liquefied Gases for Low-Temperature Food Handling. Six papers were presented here together with two commentaries. Condensed versions of several papers presented at this Symposium appear in the September issue of the JOURNAL, pages 78-83.

The Second Technical Session was convened at 9:30 a.m. by Chairman C. W. Pollock. Results of the election were reported by Executive Secretary Cross. The results are as follows:

REPORT OF INSPECTORS OF ELECTION

Total Valid Votes: 4671

	Votes received
First Vice President—John H. Fox	4669
Second Vice President—Frank H. Faust	4665
Treasurer—John E. Dube	4671
Regional Director—Region VII, Burt Lomax, Jr.	4671
Regional Director—Region I, P. K. Barker	4667
Regional Director—Region II, J. H. Ross	4671
Regional Director—Region III, E. K. Wagner	4670
Director-at-Large—P. R. Achenbach	4669
Director-at-Large—W. B. Morrison	4670
Director-at-Large—N. B. Hutcheon	4670
Director-at-Large—P. W. Wyckoff	4667

By-laws Amendments

Proposal	*For*	*Against*
1	4571	100
2	4595	76
3	4635	36
4	4645	26
5	4642	29
6	4621	50
7	4635	36
4671 Valid Votes	6 Invalid Votes	TOTAL 4677

Sherman Loud
William J. Olvany
Andrew T. Boggs, III

Inspectors

Following this announcement, four papers were presented, as scheduled.

G. F. Carlson was Chairman of the Third Technical Session, held on Wednesday morning at 9:30. Four papers were given.

Concurrent, the Domestic Refrigerator Engineering Symposium, with E. J. Von Arb as Chairman, considered the topic Quality Control. A Keynoter and three speakers participated in this session. Two papers from this Symposium were published in the JOURNAL (Mahaffay, October; Kaplan, November).

The Fourth Technical Session was held at the same time. Lincoln Bouillon was Chairman and three speakers presented papers.

The 68th Annual Meeting was adjourned by President Everetts following the General Assembly and Business Session, held at noon on Thursday.

PROGRAM—68th ANNUAL MEETING

Denver Hilton Hotel, Denver, Colo.

June 26-28, 1961

Saturday, June 24

8:00 a.m.	Advertising Committee breakfast-meeting.
9:30 a.m.	Finance Committee.
12:30 p.m.	Executive Committee luncheon-meeting.
2:00 p.m.	Standards Committee.
4:00 p.m.	Regions Central Committee dinner-meeting.

Sunday, June 25

9:30 a.m.	Program Committee.

10:00 a.m. Research and Technical Committee.
Advance Registration.
12:30 p.m. Board of Directors luncheon-dinner meeting.
2:00 p.m. International Relations Committee.
Meetings Arrangements Committee.
Membership Development Committee.
"Western Hospitality" Reception.

Monday, June 26

7:30 a.m. Speakers' Breakfasts:
First Technical Session.
Industrial Ventilation Symposium.
Air Conditioning Symposium.
Forum Moderators.
Contaminants Panel breakfast-meeting.
8:00 a.m. Public Relations Committee breakfast-meeting.
8:30 a.m. REGISTRATION.
9:00 a.m. GENERAL ASSEMBLY and BUSINESS SESSION.
Opening of the 68th Annual Meeting of the Society.
Remarks by President R. H. Tull.
Address of Welcome by Fred Janssen, Director, Region IX.
Announcement of Proposed By-laws Amendments by R. C. Cross, Executive Secretary.
New Business.
Polls Open for Election of Officers and Vote on By-laws Amendments.
9:30 a.m. FIRST TECHNICAL SESSION.
Chairman, Prof. W. E. Fontaine, School of Mech. Engrg., Dir. for Engrg. Research, Ray W. Herrick Labs., Purdue Univ.
New Development in Steam Vacuum Refrigeration—Elliot Spencer, Sales Appln. Engr., Graham Mfg. Co., Inc.
10:15 a.m. A Thermodynamic Investigation of a Refrigerant Expansion Engine—T. M. Olcott, Sr. Propulsion Engr., Convair, and Prof. H. A. Blum, Dept. Mech. Engrg., Southern Methodist Univ.
11:00 a.m. A Discussion of Some Strength Characteristics of Ice at the Interface—Profs. J. K. Stene, and W. E. Fontaine, School of Mech. Engrg., Purdue Univ.
11:45 a.m. Heat and Mass Transfer in Dehumidifying Surface Coils—Prof. Wm. L. Bryan, Dept. of Mech. Engrg., Case Inst. of Technology.
9:30 a.m. INDUSTRIAL VENTILATION SYMPOSIUM.
LABORATORY HOOD DESIGN.
Chairman, Robert Jorgensen, Buffalo Forge Co.
Basic Requirements of Laboratory Hoods—J. C. Burke, Jr., Head of Mech. Engrg., Tracy-Behrent.
Design of Hoods—E. J. Williams, Supervisor—Mech. Engrg., Abbott Laboratories.
History of the Use of Fume Hoods— G. T. Saunders, Mgr., Scientific Equip. Div., and L. N. Nelson, Head, Engrg. Dept., Kewaunee Mfg. Co.
Materials and Construction—T. B. Lanahan, Chief Engr., S. Blickman, Inc.
9:30 a.m. AIR CONDITIONING SYMPOSIUM.
WINDOW UNITS OR CENTRAL SYSTEMS?
Chairman, W. R. Moll, Sales Mgr., Air-Cond. and Dehumidifier Sales-to-Sears, Whirlpool Corp.

History, Development and Future of Air Conditioning—Frank J. Versagi, Editor, Air Conditioning, Heating & Refrigeration News.
Comfort with a Residential System—Kenneth Behr, Chief Application Engr., Lennox Industries, Inc.

12:30 p.m. WELCOME LUNCHEON.
Awards.
Announcements.
Speaker: Robert T. Person, President, Public Service Co. of Colorado.

2:00 p.m. Polls Close for Election of Officers and Vote on By-laws Amendments.

2:30 p.m. FORUMS.
Chairman, R. S. Buchanan, Asst. Dir. of Appliance Engrg. and Res., American Motors Corp.
1. Tunnels vs. Burial of Heating, Air-Conditioning and Utility Lines. Moderator: P. F. Cummings, Arch. and Engrg. Div., Minnesota State Dept. of Administration.
2. Introducing and Controlling Humidity in Heated Living Spaces. Moderator: A. G. Wilson, Head, Bldg. Services Section, Div. of Bldg. Research, National Research Council of Canada.
3. ASHRAE Research Status. Moderator: E. P. Palmatier, Carrier Res. and Dev. Co.

2:30 p.m. Exposition Committee.
Honors and Awards Committee.
Publications Committee.
Research Exhibit Committee.

4:00 p.m. RAC on System Analysis.

8:30 p.m. WESTERN PARTY.

Tuesday, June 27

7:30 a.m. Speakers' Breakfasts:
Food Refrigeration Symposium.
Second Technical Session.
Forum Moderators.

8:30 a.m. REGISTRATION.

9:00 a.m. FOOD REFRIGERATION SYMPOSIUM.
USE OF LIQUEFIED GASES FOR LOW-TEMPERATURE FOOD HANDLING.
Chairman, Dr. A. L. Brody, Prod. Dev. Mgr., M & M Candies.
The Role of Liquefied Gases in the Frozen Food Field—Prof. Kirby Hayes, Dept. of Food Technology, Univ. of Massachusetts.
Relative Merits of Various Gas Liquefying Cycles for Use in Freezing and Transportation of Foods—Prof. S. C. Collins, Dept. of Mech. Engrg., Mass. Inst. of Technology.
Effects of Ultra-Low Temperature on Foods—Dr. W. A. Maclinn, Dir., The Refrigeration Research Foundation.
Commentary: V. J. Johnson, Chief, Cryogenic Data Center, National Bur. of Standards.
Multi-Stop Delivery of Frozen Foods Using Liquid Nitrogen Refrigeration—J. J. Kane, Project Engr., Cryogenic Products Development Dept., Linde Div., Union Carbide Corp.
Use of Liquefied Gases for Freezing and Shipping of Foods—R. C. Webster, Mgr., Customer Service Lab., Air Reduction Sales Co.
Car and Truck Precooling and Post Cooling with CO_2—J. P. Antolak, Dir., Carbon Dioxide Tech. Sales, Liquid Carbonic Div., General Dynamics Corp.

Commentary: Prof. I. J. Pflug, Dept. of Food Science, Michigan State University.
TC-1.11, Insulation and Vapor Barriers.

9:30 a.m. Forum.
4. Industrial Heat Recovery.
Moderator: J. H. Clarke, Visking Company.

Second Technical Session.
Chairman, C. W. Pollock, Mgr., Air Conditioning and Refrign. Div., Crane Company.
Election Results Announced by Executive Secretary Cross.
A Study of Fluid Flow through Flexible Orifices—Prof. R. D. Ulrich, Mech. Engrg. Dept., Brigham Young University, Profs. W. E. Fontaine and O. W. Witzell, School of Mech. Engrg., Purdue Univ.

10:15 a.m. The Degradation of Polyester Film by Alcohols When Used as Additives in Refrigeration Systems—C. J. Bushouse, Chemist, General Elec. Co.

11:00 a.m. Solubility of Refrigerants 11, 21, and 22 in Organic Solvents Containing a Nitrogen Atom and in Mixtures of Liquids—Allen Thieme, 2nd Lt., US Army, Cmlc Nuclear Defense Lab. Army Chem. Center, and Prof. L. F. Albright, Dept. of Chem. Engrg., Purdue Univ.

11:45 a.m. Refrigerating Capacity and Performance Data for Various Refrigerants, Azeotropes, and Mixtures—R. C. McHarness, Div. Head, and D. D. Chapman, Engr., Freon Products Lab., E. I. du Pont de Nemours and Co.

12:45 p.m. Technical Tour of Cryogenics Laboratory.
Bureau of Standards, Boulder.

1:30 p.m. Technical Tour of United Airlines Flight Training Center and Executive Briefing Room.
Denver.

1:30 p.m. RP on Refrign. and Food Technology.

2:00 p.m. Forums.
5. Integration of Lighting and Air Conditioning.
Moderator: Walter Spiegel, Charles S. Leopold, Inc.
6. Internal Motor Protectors for Hermetic Refrigeration Compressors.
Moderator: J. E. Kumler, Ranco, Inc.
7. Problems of High Altitude Installations.
Moderator: D. D. Paxton, Bridgers and Paxton.

2:00 p.m. Education Committee.
Regional Directors Conference.
TC-1.5, Meteorology and Weather Data.
TC-6.4, Industrial Environment.
TC-7.2, Domestic Refrigerator and Unit Freezer.

3:00 p.m. RAC on Environment.
RP on Physiological Res. and Human Comfort.

6:30 p.m. Cocktail Party.

7:30 p.m. Banquet.
Installation of Officers.
Awards and Presentations.
Entertainment and Dancing.

Wednesday, June 28

7:30 a.m. Speakers' Breakfasts:
Third Technical Session.

Fourth Technical Session.
Domestic Refrig. Engrg. Symposium.

8:30 a.m. REGISTRATION.

8:30 a.m. General Administrative and Coordinating Committee.
Technical Coordinating Committee.

9:30 a.m. THIRD TECHNICAL SESSION.
Chairman, G. F. Carlson, Chief Engr., Specialty Div., Bell and Gossett Co.
Solar Heat Gains through Domed Skylights—L. F. Schutrum, Research Supvr., ASHRAE Research Lab., and Necati Ozisik, Oak Ridge Natl. Lab. Presented by C. M. Humphreys.

10:15 a.m. Air Infiltration through Revolving Doors— L. F. Schutrum, Research, Supvr., C. M. Humphreys, Asst. Dir. of Res., and J. T. Baker, Res. Engr., ASHRAE Lab., and Necati Ozisik, Oak Ridge Natl. Lab. Presented by C. M. Humphreys.

11:00 a.m. An Evaluation Procedure for Odor-Control Methods—Wm. F. Kerka, Research Engr., ASHRAE Research Lab. Presented by A. T. Boggs, III.

11:45 a.m. Daily Insolation on Surfaces Tilted Toward the Equator—Prof. B. Y. H. Liu and Dr. R. C. Jordan, Head, Dept. of Mech Engrg., Univ. of Minnesota.

9:30 a.m. DOMESTIC REFRIGERATOR ENGINEERING SYMPOSIUM.
QUALITY CONTROL.
Chairman, E. J. Von Arb, Vice Pres., Dir. of Engrg., Revco, Inc.
Keynoter: Morris Kaplan, Tech. Director, Consumers Union.
Total Quality Control—Wm. J. Masser, Mgr. of Quality Control Engrg. Service, General Electric Co.
Quality Control in Operation—Ray Sonderup, Mgr., Quality Control, Appliance Operations, Philco Corp.
Relationship of Cost Reduction vs. Quality—Wm. E. Mahaffay, Vice Pres. of Engrg., Whirlpool Corp.

9:30 a.m. FOURTH TECHNICAL SESSION.
Chairman, Lincoln Bouillon, Bouillon, Griffith, Christofferson and Schairer.
Calculated Temperature Rise in Round Ducts—J. R. Wright, Instructor, Dept. Mech. Engrg., Tennessee Polytechnic Inst., and E. J. Brown, Res. Asst. and Prof. of Mech. Engrg., Univ. of Illinois.

10:15 a.m. A Unique Hot-Box Cold-Room Facility—W. P. Brown, K. R. Solvason, and A. G. Wilson, Head, Bldg. Services Section, Div. of Bldg. Research, National Research Council of Canada.

11:00 a.m. Integrated Load Technique for Estimating Annual Energy Use of Central Air-Conditioning Plants—A. E. Congress, Head, Air Cond. and Res. Section, Power Generation Branch, Bur. of Yards and Docks, Dept. of the US Navy.

12:00 noon GENERAL ASSEMBLY and BUSINESS SESSION.
Adjournment of 68th Annual Meeting of the Society.

1:00 p.m. to 8:00 p.m. MOUNTAIN TOUR AND CHUCK WAGON.

Thursday, June 29

9:00 a.m. Board of Directors Meeting.

HOST CHAPTER
ROCKY MOUNTAIN
OFFICERS

L. R. Bindner, *President*
John Reed, *First Vice-President*
R. G. Pritchard, *Second Vice-President*
Sidney Sellers, *Secretary*
L. D. Niblack, *Treasurer*
R. J. Walker, *Board of Governors*
M. D. Beckett, *Board of Governors*
Francis Stark, *Board of Governors*
Fred Janssen, *Regional Director*

MEETING ARRANGEMENTS COMMITTEE

Fred Janssen, *Honorary Chairman*
J. H. McCabe, *Honorary Chairman*
V. J. Johnson, *General Chairman*
R. J. Walker, *Co-Chairman*
R. P. Koenig, *Finance*
J. F. Mohan, *Finance*
E. L. Heckman, *Sessions*
W. M. Richtmann, *Sessions*
G. C. Andresen, *Welcome Luncheon*
A. S. Widdowfield, *Welcome Luncheon*
W. Van Genderen, *Banquet*
George Freeland, *Banquet*
Sidney Sellers, *Entertainment*
M. D. Beckett, *Entertainment*
Paul & Irene Von Rosenberg, *Ladies*
Harry & Sally Herman, *Ladies*
J. W. Braak, *Sports*
A. W. Cooper, *Sports*
J. M. Reed, *Technical Tours*
L. R. Bindner, *Technical Tours*
L. D. Niblack, *Transportation*
L. W. Krieger, *Transportation*
J. V. Berger, *Reception*
V. E. Carrier, *Reception*
B. P. McMenamy, *Hotel*
R. L. Patton, *Hotel*
H. R. McCombs, *Publicity*
W. V. Burbank, *Publicity*
W. C. Griggs, *Printing & Signs*
J. L. Crellin, *Printing & Signs*

1751

No. 1751

New Development in Steam Vacuum Refrigeration

ELLIOT SPENCER
Associate ASHRAE

Steam vacuum refrigeration was first investigated some time prior to 1901 by LeBlanc and Parsons, and was applied quite limitedly at that time. Limitations of early ejector design and the primitive nature of pumps and controls, however, prevented its widespread use.

Despite these shortcomings, steam vacuum refrigeration systems were applied successfully during the early stages of our industry, and some of these systems are still in operation after 20 or 30 years of reliable service. During the past 25 years, improvements in accessory design and ejector efficiency have resulted in the application of these systems in increasing numbers for industrial and commercial water chilling requirements.

Typical installations of this system include both comfort air conditioning and process water cooling.

Regardless of the many known advantages of the conventional steam vacuum refrigeration cycle, which include reliability (there are no moving parts), low maintenance, the use of water exclusively as a refrigerant and the ability to use either high or low pressure steam, it was not applied as extensively to comfort air conditioning as other systems, such as centrifugal refrigeration and lithium bromide absorption. The explanation of this situation lies in the basic characteristics of steam vacuum refrigeration.

Simply stated, as the available condenser water temperature rises, both the steam rate (lb of steam per hr per ton of refrigeration) and the condenser water rate (gpm per ton of refrigeration) increase. This means that unless large quantities of condenser water are available at low cost, as from a river or other

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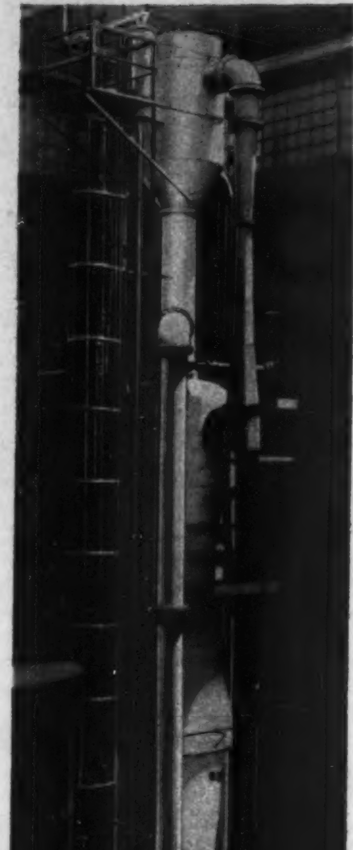
Elliot Spencer is Design Engineer with the Graham Mfg. Co. This paper was prepared for presentation at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961.

large body of water, or unless water is available at a low temperature, as from a well, cooling water requirements will substantially exceed that of the other refrigeration systems. When water from a cooling tower is used at the conventional 85 F level, steam vacuum refrigeration will have an indicated full load steam rate and water rate which is somewhat higher than either centrifugal refrigeration or lithium bromide absorption.

Typical conditions are tabulated herewith. Thus, when river or sea water is available, or some other source of cool water, steam vacuum refrigeration systems are equal, or superior, to other systems insofar as steam and water consumption are concerned. First cost is often lower. When 85 F cooling tower water is used for comfort air conditioning, chilled water temperatures of 45 to 50 F, steam vacuum refrigeration is usually ruled out. At higher chilled water temperatures, steam vacuum refrigeration will be more competitive from a utility standpoint. This can be understood better by reference to the performance characteristics of steam vacuum refrigeration systems which are tabulated and illustrated graphically herein (Fig. 5). (See Note 1.)

Obviously, any modification of the steam vacuum refrigeration cycle, which reduces its steam consumption and eliminates the need for condensing water in large quantities, will greatly affect the feasibility and usefulness of steam vacuum refrigeration for comfort air conditioning applications.



Steam - vacuum refrigeration system with barometric - type condenser

It is this modification which concerns us. By the simple expedient of replacing the water-cooled condenser of conventional steam

Completed assembly for shipment

vacuum refrigeration with a wetted air-cooled surface condenser, steam consumption is reduced and water requirements are lowered to fractional levels. Large condenser water recirculating pumps are, of course, completely eliminated.

To understand more readily how this is accomplished, it would be well to review a standard steam vacuum refrigeration cycle and to analyze some characteristic performance curves. Briefly, the conventional cycle is as follows: Fig. 1 shows how the warm water to be chilled is admitted to the flash tank (1) at the connection (2) through which it flows to the distributors (3). The vacuum on the flash tank is produced by the booster steam ejector (4) that draws off all flashed vapors and discharges these vapors to the main condenser (5). Chilled water temperature is determined by the vacuum maintained in the flash tank by the booster ejector. A vacuum of 7.63 mm is equivalent to 45 F, a vacuum of 9.2 mm is equivalent to 50 F. Propelling steam, either high or low pressure (for example as low as 2 psig), is admitted to the booster at connection (6). A higher absolute pressure (as compared to the flash tank vacuum) is maintained in the main condenser by the two stage ejector with its intercondenser (7). This ejector may be operated at steam pressures of 35 psig or higher. Higher steam pressures are pre-

Steam-vacuum system with surface-type condenser

ferred, but when only low pressure steam is available this ejector system may be replaced by a mechanical vacuum pump.

The water thus chilled by the flashing of these vapors falls to the bottom of the flash tank and is removed by a hotwell type centrifugal pump (8). The cooling water for circulation to the main condenser is admitted at Point (9), as indicated. The level in the flash tank is maintained by the liquid level controller (10). From the foregoing, it is apparent that the system is extremely simple; that there are no moving parts other than the centrifugal chilled water pump.

In Fig. (1A), the surface condenser (5) has been replaced with a barometric condenser. Barometric condenser systems operate at lower condensing temperatures and show correspondingly better steam and water rates than do surface condenser systems.

In a barometric condenser, steam is condensed by direct contact with water. This means that the condensing temperature can economically approach within 5 F of the leaving condenser water temperature. In surface condensers, however, the minimum economical approach is 10 to 15 F. In a surface condenser we have the added heat transfer resistance brought about by the possibility of fouling the surface. In a barometric condenser there are no heat exchange surfaces, and therefore fouling is no problem. Maintenance is corre-

spondingly reduced. Barometric systems will not fall off in capacity or increase their steam consumption with time due to the fouling of the condenser tubes, as will any surface condenser system. This applies to absorption as well as centrifugal refrigeration systems using tubular heat exchange surfaces.

Steam vacuum refrigeration systems using barometric condensers are not usually applied to comfort air conditioning systems, because of space limitations and the requirement for a 34′ barometric leg. It is probable that in many cases additional investigation on the part of the design engineer of such systems would indicate the possibility of installing barometric densers, in view of their advantageous characteristics. In some cases barometric condensers have been installed within a building.

However, it can be stated that in the majority of cases comfort air conditioning systems cannot use barometric condensers, and, therefore, under most circumstances this specific steam vacuum refrigeration cycle is not used.

For comparison purposes, we will examine the characteristics of the system in Fig. 1 under full load design conditions shown in the diagram. These conditions are given in Table I.

Table I

Chilled Water Temperature	45 F
Condenser Water Temperature	85 F
Wetbulb Temperature	78 F
Drybulb Temperature	95 F
Steam Pressure	100 psig
Condenser Water Range	10 F

Such a system would use (at full load) 25 lb/hr of steam per ton of refrigeration and 6.6 gal/min of water per ton of refrigeration. For this condition the condensing temperature would be 103 F.

Steam consumption and water requirements are given at full load and at the design wetbulb condition. The presumption is that water is coming from a cooling tower at 85 F at the design wetbulb condition. As the wetbulb drops, the water which comes from the cooling tower will be cooler and the steam consumption of the steam vacuum refrigeration system will be reduced.

Comparable characteristics for lithium bromide absorption and steam driven centrifugals under full load design conditions would be similar to those shown in Table II below.

In order to compete with other steam-operated refrigeration systems, steam vacuum refrigeration steam and water rates must be reduced.

Table II

	Lithium Bromide Absorption	Steam Driven Centrifugal Refrigeration
Steam	19.6 pph/T.R.	16 pph/T.R.
Condenser Water	3.5-4 gpm/T.R.	3 gpm/T.R.
Condenser Water Rise	16 F	20 F
Return Condenser Water Temp.	101 F	105 F

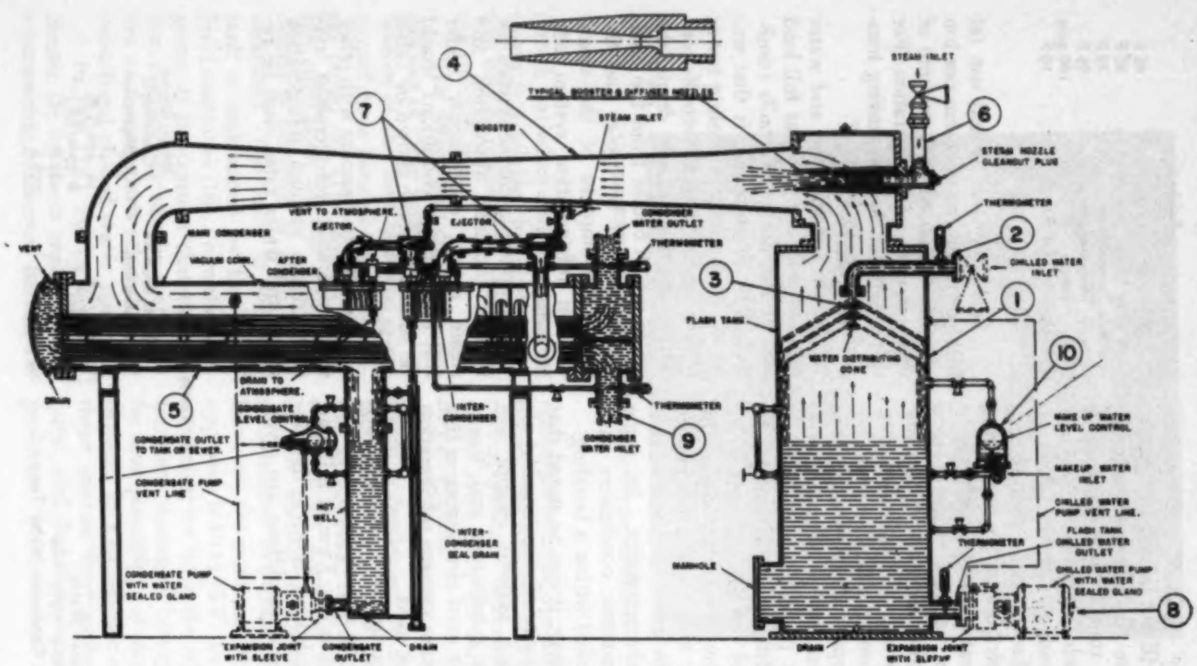


Fig. 1 Steam vacuum refrigeration system with surface type condenser. 1—Flash Tank. 2—Chilled Water Inlet. 3—Distributors. 4—Booster Steam Ejector. 5—Main Condenser. 6—Steam Inlet. 7—Intercondenser. 8—Centrifugal Pump. 9—Condenser Water Inlet. 10—Liquid Level Controller.

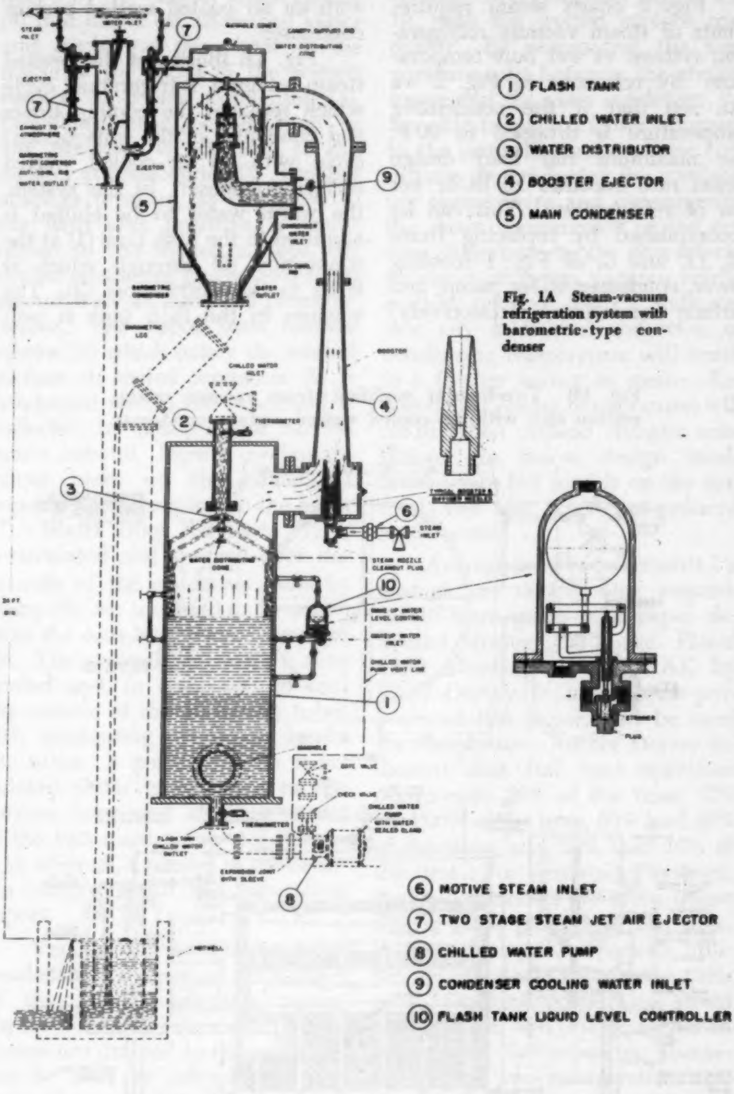
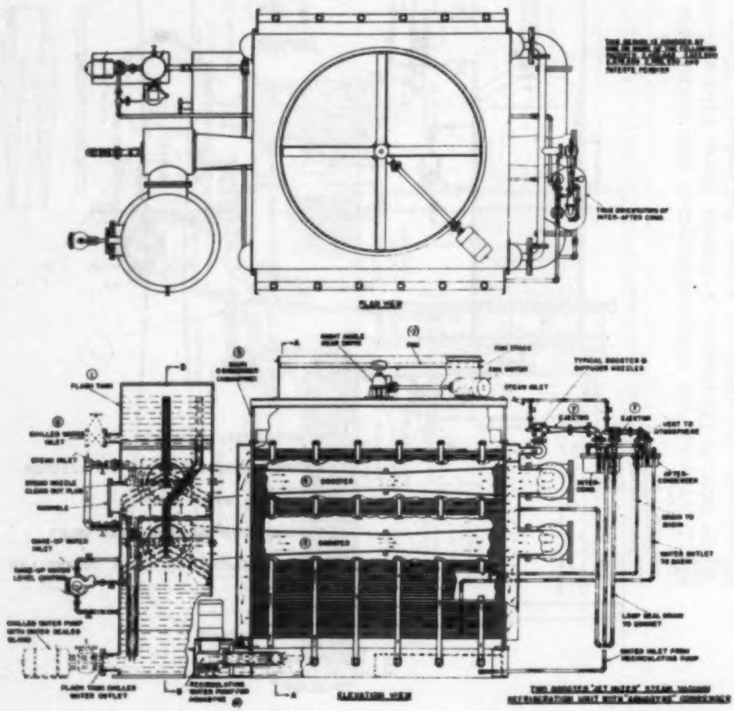


Fig. 1A Steam-vacuum refrigeration system with barometric-type condenser

Fig. 2 charts steam requirements of steam vacuum refrigeration systems vs wet bulb temperature. By reference to Fig. 2 we can see that if the condensing temperature is dropped to 90 F, the maximum full load design steam rate becomes 16 lb/hr per ton of refrigeration. This can be accomplished by replacing Items 12, 11, and 5, of Fig. 1 (cooling tower, condenser water pump, and surface condenser, respectively) with an air cooled wetted surface condenser.

Fig. 1B illustrates the revised steam vacuum refrigeration cycle which replaces the cooling tower and condenser of the conventional cycle with an air-cooled wetted surface condenser. In this system, the warm water to be chilled is admitted to the flash tank (1) at the connection (2) through which it flows to the distributors (3). The vacuum in the flash tank is pro-

Fig. 1B Two-booster modified steam vacuum refrigeration unit with air-cooled wetted surface condenser



duced by the booster steam ejector (4) that draws off all flashed vapor and discharges this vapor to the inside of the tubes of the wetted surface air-cooled condenser (5). Propelling steam, either high or low pressure, is admitted to the booster at connection (6). A higher absolute pressure, as compared to the flash tank pressure, is maintained in the wetted surface air-cooled condenser by the two stage ejector with its inter and aftercondenser. The vapor from booster ejector (4) which enters the wetted surface air-cooled condenser (5) is condensed within the tubes and is collected by headers and carried into a hotwell. Here a condensate pump picks up the condensed vapor and discharges it to the basin (7). Water from the basin (7) is recirculated and sprayed over the outside of the condenser coils by pump (8). Air is caused to be drawn over the coils by the condenser fan (9). The recirculated water is thus cooled and, in turn, acts to cool the outside of the condenser tubes (10), condensing the vapors inside the tubes. A portion of the recirculated water is bypassed to the surface inter and aftercondensers of the two stage air ejector system (11), wherein it is used to condense the motive steam and saturation vapors.

Vapors condensed in the intercondenser are piped to the suction of the main condensate pump. vapors condensed in the aftercondenser are drained to the basin. As can be seen by reference to the cycle shown in Fig. 1B, we have replaced the cooling tower and cooling tower pumps of Figs. 1 and 1A with the wetted surface condenser. By eliminating the temperature split between the cooling tower and the main condenser, we are able to make a closer approach to the prevailing atmospheric conditions, thus reducing the condensing temperature and consequently the steam consumption of the system. Reference to the characteristic curve of steam vacuum refrigeration systems, Fig. 2 indicates that any additional reduction of condensing temperature will result in a further saving in steam. Reduced condensing temperatures will result when outside climatic conditions are below design levels (even when full load is on the system), and also whenever reduced loads occur.

Average load requirements for seasonal air conditioning requirements were given in a paper delivered during the Lake Placid 1959 Meeting of ASHRAE by W. G. Dorsey, Jr., and, for the purposes of this paper, will be used for comparison. Author Dorsey indicated that full load operation occurs only 20% of the time, 75% load 30% of the time, 50% load 40% of the time, and 25% load 10% of the time. For year round systems, load requirements are often below 25% a large portion of the time. Further, during these periods, utilities selling steam often raise their price. Obviously, part load steam consumption is a major factor in determining the economic characteristics of any refrigeration system. To take the total number of hours of operation at each percent-

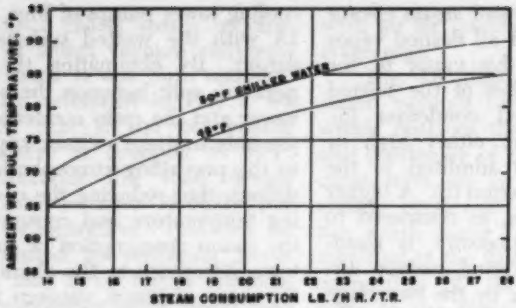


FIG. 2

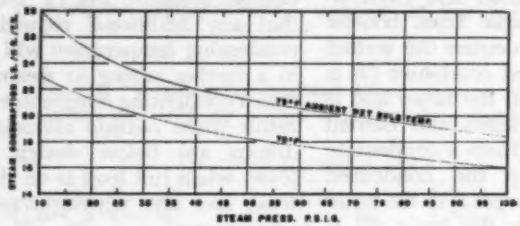


FIG. 3

Performance charts for steam-vacuum refrigeration systems vs. wet bulb temperatures

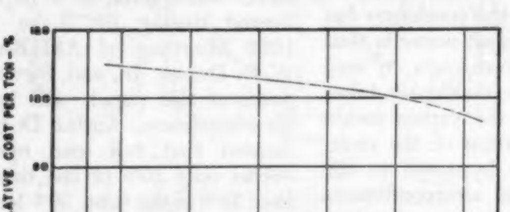


FIG. 4

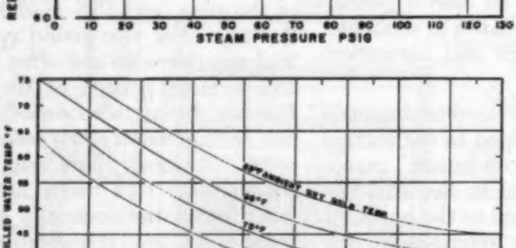


FIG. 5

age load and add them to obtain equivalent full load hours of operation is inaccurate. Since steam vacuum refrigeration, centrifugal steam-driven refrigeration and lithium bromide absorption operate at varying efficiency at part load levels, careful evaluation of loads and outside climate conditions and their relationships to the loads should be made in any complete analysis. The modified steam vacuum refrigeration cycle has performance characteristics which are comparable to that of steam driven centrifugal systems at full load and at partial loads. This system requires less steam than lithium bromide systems, and has eliminated the need for an external cooling water source.

The effect of steam pressure on any steam vacuum refrigeration cycle can be seen by reference to the curve of Fig. 3. Steam consumption increases as steam pressure is decreased. Regardless of this situation, by use of the wetted surface air-cooled condenser, steam vacuum refrigeration can now be designed to have maximum full load steam consumptions of 20 lb/hr per ton of refrigeration, even with steam available at 15 psig. This is equal to, or better than, the steam consumptions to be expected from absorption or steam driven centrifugal systems. At reduced loads, of course, steam consumption will be reduced as indicated earlier. Steam vacuum refrigeration systems designed for lower pressure steam with wetted surface air-cooled condensers, will be somewhat more expensive than systems designed for higher pressure steam and will use more steam. The use of the modified steam vacuum refrigeration cycle will be adversely affected by reduced steam pressure.

All steam vacuum refrigeration systems respond to increased chilled water temperature with improved efficiency and the modified cycle is no exception. Higher chilled water temperature means not only reduced steam consumption at all load conditions, but also reduced first cost due to decreases in equipment size. At higher chilled water temperatures, the modified system becomes smaller and lighter in weight.

Fig. 5 shows how chilled water temperature affects steam consumption (see Notes 1 and 6). Since chilled water temperature affects steam consumption and equipment size, it is sometimes worthwhile to consider a two stage flash system when the chilled water temperature rise is higher than the conventional 10 F, for example 20 F as used on high rise coils. In such systems the returning chilled water is first cooled from say 65 to 55 F in one flash chamber, and then from 55 to 45 F in the second flash chamber. Approximately half of the refrigeration load is chilled at the 55 F level. Reference to Fig. 5 indicates that the lb of steam per hour per ton of refrigeration required for this portion of the load is less than that required for that portion of the load done at the 45 F level. The overall steam consumption will be lower than if all of the work were done at the final flash temperature of 45 F. Generally

Table III—Comparison Chart of Various Steam Powered Refrigeration Systems

200 T. R. Systems Selected for 45 F Chilled Water—100 psig Steam—85 F Condenser Water —78-W.B. (see Note 6)

	Modified Steam Vacuum Refrig.		Lithium Bromide Absorption Refrig.		Steam Driven Centrifugal Refrig.	
External cond. water—gpm	No external cooling water required.		3.5 to 4		3	
Cond. water temp. rise	——————		16 F		20 F	
Effect of reduction in cond. temp. below design conds.	5 F drop produces 10% reduction in steam flow.		No steam saving. Drop in cond. temp. unbalances machine. Cond. temp. is held constant. With solution control steam need is reduced at part load. (See below).		Vacuum on steam cond. increases, increasing efficiency of turbine. Max effect approx. 7%. At low cond. temp. operational difficulties result.	
Steam Consumption	Equiv. Steam Rate	% Full Load Steam	Equiv. Steam Rate	% Full Load Steam	Equiv. Steam Rate	% Full Load Steam
Full Load	16 lb/hr/T.R.	100	19.5 lb/hr/T.R.	100	16 lb/hr/T.R.	100*
75%	14	65.7	20.8	80	14.45	68
50%	12	37.6	23.4	60	13.75	43
25%	10	15.60	34.3	44	19.2	50
			(without solution control)			
	(Figures given are at reduced wetbulb at part load conditions. See Note 6.)		Equiv. Steam Rate	% Full Load Steam	* For units having larger capacities, rates as low as 13 lb/T.R. are possible.	
			19.5 lb/hr/T.R.	100		
			18.5	80		
			17.7	60		
			17.4	40		
			16.8	10		
			(with solution control)			
Cond. Water Pump hp	2-hp recirculating spray pump on evaporative condenser.		40		20	
Auxiliary hp	8½ approx. 1 lb/hr of steam will be used for each T.R. capacity for secondary ejectors.		10		5/6 (intermittent for purge unit) approx. 0.75 lb/hr of steam will be used for each T.R. for ejector on steam cond.	
Cooling Tower or Mod. Steam Vac. Fan hp	20 (Fan hp)		20		15	
Water Make-up gpm	33 gpm (includes blowdown).		90 gpm (includes blowdown)		66 gpm (includes blowdown)	
Licensed engineer required by local code	No (see Note A).		No		Yes	
Maintenance Required	Maint. fans and pumps.		Clean cond. and absorber. Maint. fans on tower & cond. water pump. Special treatment to protect against corrosion during winter.		Maintain cooling tower fan, cond. water pumps, centrifugal compressor and steam turbine.	

Table III — Continued

	Modified Steam Vacuum Refrig.	Lithium Bromide Absorption Refri.	Steam-Driven Centrifugal Refrig.
Space Required	Entire system uses approx. same area as cooling tower for conventional systems. Entire unit installed outdoors.	Approx. 10% more cooling tower area than steam driven centrifugals plus indoor space of 5 x 19 x 8 ft high.	Approx. 10% less than absorption for cooling tower plus 8 x 15 x 7 ft high indoor space.
Weight	Total system weight is greater than cooling tower alone but less than total weight of conventional systems & tower.	Tower 10% heavier than centrifugal. System about 25% lighter than centrifugal.	Tower 10% lighter than absorption. System about 25% heavier than absorption.
Noise & Vibration	Low	Low	Low
Enclosed engine room usually required	No	Yes	Yes
Cond. water required	No (make-up for Evap. Cond. only)	Yes (plus make-up for tower)	Yes (plus make-up for tower)
Cost % of piping external to tower or Aquamizer	(See Note B)	100% plus	100%
Cond. water pump required	No (small pump included in system cost)	Yes	Yes
Relative Cost	0	100 plus	100
Cooling tower required	No	Yes	Yes
Relative Cost	0	100 plus	100
Refrigerant	Water	Water (Lithium Bromide Absorbant)	Refrigerant
Control	Automatic 0-100%	Automatic 0-100%	Automatic start-up manual
Relative installed first cost includ. cooling tower, steam piping, cond. water pumps and cond. water piping. Starters on all motors included	Entire unit on roof-outdoors. 80%	100 plus 10%. Tower on roof. Refrig. unit in basement. 100%. Tower on roof. Refrig. unit on roof in engine room. (Cost of engine room not included.)	100. Tower on roof. Refrig. unit in basement. 95. Entire system on roof. Refrig. unit indoors. (Cost of engine room not included.)

Note A: Local Codes vary but usually require Licensed Engineer for halocarbon systems. For both absorption and steam vacuum refrigeration no engineer is required when low pressure steam is used. For steam vacuum refrigeration when high pressure steam is used, Licensed Engineer may or may not be required, but when steam is supplied by local utility and no high pressure boiler is maintained on premises, Licensed Engineer is not a requirement.

Note B: No external cooling water piping is required but when unit is remote from steam source and competing systems would normally be located in basement, added cost of steam piping to unit must be considered.

speaking, the response of steam vacuum refrigeration systems to higher chilled water temperatures is greater, proportionately, than the response of either absorption or centrifugal refrigeration systems.

Steam vacuum refrigeration systems may be hand-controlled, partially automatic, or fully automatic. The simplest way to regulate the capacity of steam vacuum refrigeration systems is to install several ejectors in parallel and operate only as many ejectors as are required to handle the heat load. A simple automatic on-off type of control is used for this purpose. By sensing the chilled water temperature out of the flash tank, the controller can turn the steam on or off to each ejector in consecutive order to vary the refrigeration capacity of the system.

When multiple boosters are used, the flash tank is divided into compartments (see Note 2); each compartment operating in conjunction with its own booster. As the load changes, either in the nature of an increase or a decrease, a booster will be turned on or off and the water valve of that compartment will be opened or closed to synchronize with the corresponding steam valve. This may be done either through the use of automatic controls or by hand. Each compartment is portioned off by a double baffle that extends to within about 12 in. of the bottom of the flash tank. This opening at the bottom, joining all compartments together, is always submerged.

When a booster is shut off, the compartment it services is then subject to a rise in pressure which causes the water level to depress, whereas the water level rises in the compartments that are under the higher vacuum of the operative booster. The flash tank is sufficiently deep to allow for these variations in the water level and still maintain a seal between compartments (see Note 2). Make-up water is admitted through a float controller that is regulated by the level in the compartments that are in operation. Failure of controls cannot damage the steam vacuum refrigeration unit, nor can faulty operation of the control system result in damage. Damage from freeze-up is not possible.

Table III is a comparison chart of the various steam powered refrigeration systems. Figures given are based upon the design rates shown therein. Although the modified steam vacuum system selected for this comparison shows a steam rate and first cost which favor it over both steam driven centrifugal and absorption systems, the steam vacuum system could be designed for a higher steam rate at a corresponding reduction in first cost. Under these conditions, the steam vacuum system would still be competitive in steam consumption with absorption systems. Thus. when only low pressure steam is available, a steam vacuum system may be selected for a peak rate slightly higher than the design peak rate for absorption. Part load rates, taking reduced wetbulb into consideration, are still lower than absorp-

tion (see Note 3), which leaves the steam vacuum system still at an advantage insofar as steam consumption is concerned.

The response of the modified steam vacuum system to lower condensing temperatures is a most important characteristic. A recent investigation of published weather data for a northern city indicated that out of 976 hr of an operating season, only 3 hr averaged above the wetbulb design condition, and only 54 hr were above the 70 F wetbulb condition; 440 hr were below 66 F wetbulb. The prevalence of wetbulb conditions at or above the design point will be influenced by the location of the specific installation. However, the proportion of time during which a refrigeration system operates at or near design conditions, insofar as wetbulb is concerned, is quite small, and any system which benefits substantially when these conditions are reduced offers a very important advantage. The resultant savings in steam consumption at lower wetbulb conditions make the modified steam vacuum system most attractive from an operating economy standpoint.

With no moving parts and only water as the refrigerant, maintenance and repairs are quite low. Records for even the earliest steam vacuum refrigeration systems indicate that repair bills averaged $20 per year. Life expectancy has been excellent, far greater than that of machines with moving parts. Maintenance experience with the new steam vacuum system has been too limited to be accurately predicted. However, since the low side of the system is no different than previous steam vacum refrigeration systems, it would be expected that maintenance experience on this part of the system would duplicate that on previous systems.

It is anticipated that maintenance on the high side (condenser) of this system would be approximately equivalent to that on a cooling tower. Maintenance of fans and recirculating water pumps is required for both. The recirculating water pump of the modified steam vacuum system would be smaller than the main condenser water pump of the cooling tower; however, the addition of the small condensate pump on the modified steam vacuum system would make up for this size differential. We believe that the overall maintenance requirements for the modified steam vacuum system will be less than the requirements for any other system plus its cooling tower.

One special advantage of the modified steam vacuum system is that the "fill" in the condenser is non-flammable, which means there is no off-season fire hazard such as there could be for wood fill cooling towers. The absence of moving parts on the low side of the system means there will be no vibration at either the high or low frequency level. Noise levels are also low, less than that of a fan and water on a cooling tower. There is no "hunting" noise such as can occur on other systems operating at extreme low loads.

The cost of installing any system is hard to evaluate, depending

upon where the unit is installed, whether it is located on the roof or in the basement. The steam vacuum system is usually installed outdoors; in the case of a tall building it is installed on the roof where the cooling tower usually would be installed. The rigging cost of the modified steam vacuum system is usually less than, or about equal to, the rigging cost of a cooling tower alone. If it is planned to install the entire refrigeration system on the roof or in a penthouse, then rigging costs of the other systems would differ little from the rigging costs of a modified steam vacuum system. When it is planned to install the other refrigeration systems in the basement and the cooling tower on the roof, the combined rigging costs of the two procedures should be larger than that for the modified steam vacuum system installed on the roof. Since the steam vacuum system is customarily installed out of doors, it would require no engine room. In addition to the saving brought about by elimination of the engine room, rigging should be simplified due to the elimination of encumbrances produced by walls, etc. The cost of condenser water piping becomes a major factor when comparison is made between a modified steam vacuum system installed on the roof and any other system installed in the basement with a cooling tower on the roof. The cost of steam piping will be increased since a steam line must be run from the street to the roof.

Table III provides relative costs of the various refrigeration systems. Investigation has shown that the overall first cost of the modified steam vacuum system is lower than that for other systems, with the possible exception of electric-driven centrifugal systems when these systems are installed in penthouses immediately adjacent to the cooling tower. In arriving at these evaluations, the elimination of protective housing for the steam vacuum system has not been taken into consideration. This saving, especially in existent buildings, will be quite advantageous to the steam vacuum system. At higher chilled water temperatures, the modified steam vacuum system will prove even more attractive than at the temperatures considered.

The modified steam vacuum system occupies approximately the same space normally required for the cooling tower alone for conventional steam-operated systems. Dimensions for a typical 225 ton, 45 F, modified steam vacuum system approximate those for a 300-ton, 50 F chilled water system. Moreover, space requirements for larger tonnages would not necessarily increase proportionately. Since the entire system is installed where the cooling tower alone would have been installed, machine room space is saved for other uses.

Weight requirements of the modified steam vacuum system are greater than the weight of the cooling tower alone for steam-operated refrigeration systems, but the weight of the entire system is about equal to, or slightly less than, the combined weight of a cooling tower and steam-operated refrig-

eration system where the entire system is installed on the roof. Where it is required to locate the modified steam vacum system on a lower floor set-back, the system would have the disadvantage of losing some of the available chilled water pumping head. This occurs when the returning chilled water is flashed to the vacuum in the evaporator. Closed chilled water cycles available in other systems do not have this disadvantage. In both lithium bromide absorption and centrifugal refrigeration, water is chilled under the operating static head pressure, but in the modified steam vacuum system the water must be flashed to the vacuum in the evaporator, resulting in a static head loss and consequent greater pumping horsepower. This is not the condition when the entire system is located on the roof.

Although the modified steam vacuum system operates under a vacuum (see Note 4), this vacuum exists only up to the discharge side of the chilled water pump. Water returning to the evaporator is under a positive pressure. This means that the only air leaks that can occur must take place within the system itself. Proper design and minimization of joints through which leakage can occur have made air leakage a negligible problem in these systems.

As a further precaution, vacuum is maintained in these systems by a large two stage ejector system with air handling capacities much greater than that found on either absorption or centrifugal systems. The danger of air leakage in steam vacuum refrigeration systems is only slightly greater than the danger of air leakage in the surface condenser and turbine combination of a centrifugal refrigeration system. Experience has indicated that this problem is practically non-existent, except during the original start-up of the system (see Note 5).

In summary, a new steam-operated refrigeration system is now available for comfort air conditioning; a system with low installed first cost, low annual steam consumption, and one which is reliable and maintenance free. Although limited in size at the present time to systems under 500 ton of refrigeration in a single package, the merits of this modified steam vacuum system should be considered wherever steam is used to produce refrigeration for air conditioning or other water chilling purposes.

Note 1: Characteristic curves (Fig. 5) are for a series of boosters, each designed for the specific conditions shown on the curve.

Note 2: Description of part-load control is applicable to the systems shown in Figs. 1 and 1B. The modified steam vacuum system and the vertical flash tank systems shown in the illustration use a stacked flash tank eliminating the need for shutting water valves as boosters are turned on and off and also, incidentally, eliminating any inefficiency which would result from water stored in the now operating flash tank.

Note 3: Absorption refrigeration part-load rates have recently been improved by a technique known as solution control. An accurate representation of part-load performance of any given ejector system cannot be obtained by reference to Fig. 5, which represents the performance of a series of boosters; each designed for the specific conditions shown on the curves. A characteristic throttling curve would more accurately depict performance for any system selected. Thus, performance of a system selected for operation at 90 F condensing temperature would not be

as good as shown in Fig. 5 when this system operated at 90 F condensing temperature; i.e. the throttling curve would not be a straight line. The figures given, however, are accurate enough for the purpose of this article.

Note 4: Early steam vacuum refrigeration systems were troubled by loss of vacuum due to poor piping techniques and insufficient ejector capacity. Modern systems have ejectors with capacities many times those of earlier systems and several times the capacity of the steam ejectors used on the surface condensers of all steam-driven centrifugal systems. By designing the system so all piping is under a positive pressure, leakage problems can be limited to the steam vacuum refrigeration unit itself. Present day steam vacuum refrigeration systems encounter no difficulty with maintenance of vacuum.

Note 5: Breaking.

Loss of vacuum and insufficient condenser capacity were also responsible in early systems for an effect known as breaking.

When a booster ejector breaks, the motive steam discharges to the flash tank resulting in a heating rather than a cooling process. Breaking occurs when the compression range over which the ejector is required to function exceeds that for which it was designed. This can occur if the condensing temperature is higher than design.

Excessive condensing temperature can be eliminated by simply designing the evaporative condenser portion of the unit with sufficient excess surface to anticipate the worst conditions found in local weather records for the installation in question.

An over-riding control device can be incorporated in the system which will permit the chilled water temperature to rise a few degrees during those few hours when outside W.B. conditions exceed the maximum which was anticipated during design. This rise in chilled water temperature will decrease the range over which the ejector must compress and eliminates any danger of breaking.

Proper design of condenser and ejector with a 15% safety factor in condensing surface and a booster designed to operate without breaking at condensing temperatures 2-3 F above design will not only insure against breaking but will improve performance at all load conditions since excess surface will result in reduced condensing temperature at all loads.

Breaking can only occur at full load when condensing conditions exceed design, or if load is greater than design. This can be accommodated by the steps described above at all other times; i.e., when load is less than design or when W.B. is less than design, there is no possible danger of breaking.

Breaking can only occur on a poorly designed system.

Note 6: As load is reduced, condensing temperatures are reduced, since at reduced load evaporative condenser is oversized and closer approach to wetbulb occurs—at 100% W.B. has assumed 78 F at 75% load W.B. is 73 F at 50% of load W.B. is 70 F at 25% load W.B. is 64.5 F. It must be emphasized that any comparison of the relative energy requirements of one system, as opposed to another, must be made by a detailed analysis of the variation of load with wetbulb. Load does not necessarily drop off as the wetbulb is reduced. Internal load proportions, fresh air requirements, etc., will make a great difference. Analysis must be made for each specific case. A 25% load at 78 W.B. would require more steam per T.R. than a 25% load at 70 W.B. The above figures however do reflect the dramatic reduction in steam requirements as load and W.B. decrease.

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y. (Written): The refrigeration system which is described by the author is an interesting and significant application of the evaporative condenser to the improvement of the steam ejector type of refrigeration equipment. It is of historical interest that what is believed to be the first application of the forced circulation evaporative condenser was to a steam ejector refrigeration system. Four thousand of these were applied for the air conditioning of railroad cars largely in the early and middle 30's.

Unfortunately, the present paper leaves many important questions unanswered. The application of steam jet boosters to supply refrigeration for air conditioning has been limited in the past, not only because of the relatively high steam rate but also by the tendency of the ejector to "break" when the condenser pressure became too high or the steam pressure dropped too low. When this occurred, the system became a heating system instead of a cooling system. Unfortunately, it occurred at the time when the wet bulb temperatures were at a maximum and when the system was most needed for cooling.

While it is true that the use of the evaporative condenser has decreased the danger of high condenser temperatures due to fouling of the condenser surfaces, it is still true that wet bulb temperatures can rise substantially above the design wet bulb. I well remember a day in Philadelphia when I was checking one of our commercial steam jet systems, when the wet bulb temperature was measured at 85 deg at the equipment intake at the top of the building. Because of this characteristic, our equipment was designed to operate at wet bulb temperatures close to 90 F. My first question to the author, therefore, is what margin of safety is the equipment described in his paper designed for. Specifically, will a system designed for operation with a 78 deg wet bulb operate at full load at 85 deg wet bulb and at design steam pressure.

Quite a point is made in the paper of the reduction in steam rate with reduced wet bulb temperatures and capacities. It should be understood that the steam jet is essentially a constant steam flow device at a given steam pressure. Thus, in order to get lower steam rates at the same chilled water temperature, it is necessary to throttle the steam flow, usually by reducing the steam pressure. It would be helpful if the

author would tell us how this control is effected, and if it is done automatically, whether the steam rates shown on Figs. 2, 3, and 5 were obtained using this automatic control.

In the text and in Table III the author gives a steam rate of 16 lb/hr/ton of refrigeration at full load, and according to Note 6 this is for a designed wet bulb of 78 deg; but in Figs. 2 and 5, he shows a figure for 45 deg chilled water temperature, 78 deg wet bulb temperature of slightly above 20 lb/hr/ton—which of these figures is correct?

The author gives a maximum full-load steam consumption of 20 lb/hr/ton of refrigeration with steam available at 15 psig; yet in Fig. 3, he shows a steam rate of 22 lb/hr/ton at 70 deg ambient wet bulb and more than 26 at 75 deg ambient wet bulb—which of these is correct?

Unfortunately, the author does not give rating information in terms of the usual basis of comparison using condensing temperature vs. chilled water temperature. The latest issue of the ASRE Design Data Book gives representative steam rates of 20 lb/ton for 45 deg chilled water and 92 deg condenser, and 16 lb/ton for 85 deg condenser. Both of these figures appear to be remarkably close to the 78 deg wet bulb temperature rating. It would be helpful if the author would indicate what the design condenser temperature is with 78 deg wet bulb, and also, to extend the curves in Fig. 5 to show the performance at lower wet bulbs and steam consumptions shown in Table III.

In Table III, the author shows a steam rate of 12 lb/hr/ton, 50% cooling capacity and Note 6 indicates that this is at a wet bulb temperature of 70 F. However, for 45 deg chilled water and 70 F ambient wet bulb, Fig. 5 shows a steam consumption of 16 lb/hr/ton—which of these figures is correct?

The curves in Fig. 3 are indicated as being for a series of ejectors designed for the specific steam pressures shown rather than for a single ejector; thus they do not represent the performance which would be obtained by the throttling of an ejector designed for a higher pressure. I would like to ask the author whether the steam consumption shown in Fig. 2 and Fig. 5 are also for a line of ejectors or whether they represent the throttled performance of a single ejector. The reason why this question is important is that the throttled performance is poorer than the performance of an ejector designed for this specific condition.

In considering the comparison of part-load performance shown on Table III, several factors must be taken into account. First, the steam jet figures are for an "open" type system in which the water flashed in the evaporator is circulated through the cooling coils in the air conditioning system. While undoubtedly such a system can be made to work successfully on a close coupled basis, our experience with it, both with steam jet and absorption refrigeration has been so adverse that we no longer use it. Pertinent to this point, it would be helpful if the author could tell us what the introduction of an indirect cooling system would do to the steam rate.

The second point is that the steam jet figures are for 100 psig steam while the figures quoted for lithium bromide absorption refrigeration are actually for 15 psig steam. This offers several additional possibilities of saving for the absorption type system, such as the use of the excess pressure in the steam for the driving of auxiliaries or a parallel steam driven centrifugal refrigerating unit. If the steam is used direct, there is somewhat more Btu/lb available at 100 than at 15 psig. Furthermore, the condensate available at approximately 240 F from the absorption machine can be used for heat exchange either within the system or for hot water heating.

The next point is that the figures shown do not include the steam used for purging, which would increase the rates to 17, 15.3, 14 and 14 lb/hr/ton respectively for 100, 75, 50 and 25% respectively. There is no corresponding use of steam in the absorption machine.

Perhaps the most important difference is that whereas the absorption control is a modulating one which works throughout the entire capacity range, the control of refrigeration capacity in the steam jet must be by turning off and on ejectors. Presumably, the figures given are for a four-ejector system, each percentage being at the maximum capacity of the ejectors being used. It would be helpful if the author could give us figures for the performance at other capacities. In making such a comparison, it should be kept in mind that the refrigerating machines should pull down to lower temperatures within the range of the off and on control points of the thermostat. Furthermore, each time an ejector is turned off, steam at condenser pressure and temperature enters the evaporator compartment of this ejector and heats the water and metal to the condenser temperature. The more frequently the control operates, the greater is the loss from this source. Could the author tell us what the magnitude of this loss is?

The author comments on the low noise level of the equipment. While our evaluation of noise was not as refined in the days of my experience with steam jets as it is today, my recollection was that our ejectors were quite noisy and with a rather unpleasant quality. Since it is proposed to use the machine out in the open, the noise problem could be one of real concern. Can the author give us any specific information on the sound power level output on a frequency band basis or if this is not available, on the sound pressure level output on a frequency band basis?

I feel sure that a clarification of some of the questions raised above will be helpful not only to me, but to all of the members of our

Society who are interested in new developments in refrigeration and will help us to evaluate this new machine realistically in relation to other types which are available.

AUTHOR SPENCER: Mr. Ashley comments that early application of evaporative condensers to steam vacuum refrigeration was made in the middle 30's, and that experience with the breaking of the ejectors caused his company to abandon further investigation. The application referred to by Mr. Ashley was that of railroad car air conditioning, where similar difficulties with evaporative condensers occurred when refrigeration systems using these devices were installed. I believe that some of the difficulties encountered were peculiar to the application itself with its inherent dirt pickup, wet bulb variations, and space limitations.

Mr. Ashley has gone into great detail concerning the tendency of steam vacuum refrigeration systems to break. I believe that this question was answered in sufficient detail under Note 5 of this article. The equipment, as we design it, is not designed to operate at full capacity without breaking when the wet bulb temperature exceeds design, as in the case of the example by some 7 deg, namely 85 vs. 78 deg wet bulb. Under similar conditions cooling towers for normal applications would also fail to produce design water temperatures, and no attempt is made to design these devices for such a wide variation in wet bulb temperatures above those for normal design. It is our custom to design our equipment with 15% excess surface, and it would be possible to prevent breaking of the ejectors by the steps outlined in Note 5.

Mr. Ashley raises several pertinent questions with regard to the performance described in the text and the performance shown on the curves, and comments that the curves seem to agree closely with data shown in the ASRE Data Book, but do not compare with the stated performance in the text of the article. Both the curves and the article are correct. The performance of these systems is influenced by the amount of surface which is incorporated in the evaporative condenser. An increase in surface results in a closer approach to the wet bulb condition. The performance as described in the text is for a unit designed for close wet bulb approaches. The performance as shown in the curves is for standard wet bulb approaches. It was felt that use of these curves in the future by people evaluating a given problem with regard to the possible use of a system such as that described in this article would be "safe", but the text would indicate additional economies which could be accomplished by increasing the amount of surface in the evaporative condenser.

The curves for Figs. 2 and 5 represent the anticipated throttled performance of a single ejector, although the precise performance of any individual ejector will depend upon the specific design conditions for which it, and the system, are selected.

Mr. Ashley has commented with regard to the fact that the figures given are for an open system, and he asks what the introduction of indirect cooling would do to the steam rate. The answer to this question is obvious. The steam rate would be drastically increased if an indirect system were applied. It must be pointed out, however, that there are hundreds of steam vacuum refrigeration systems now installed which use the direct flash method without difficulty. Reference to Note 4 should be made.

There is no argument with Mr. Ashley's comment that the absorption system could use 100 lb motive steam for driving auxiliary equipment before discharging it at 15 psig for use in the absorption system, and there is no argument that the higher condensate temperature means that where condensate is returned economies will be realized, and there is certainly no argument with the fact that systems using compound absorption refrigeration and centrifugal refrigeration offer very real steam economies, but for other reasons which we shall not go into here it is often not practical, nor economical, to accomplish the steam savings inherent in running rotating equipment with steam before its latent heat is used in an absorption system.

Mr. Ashley's comment with regard to the type of control used on steam vacuum refrigeration systems is well taken, and there is no question that as the ejectors are turned on and off losses occur due to heat-up of the evaporator compartment by vapor being flashed from the barometric condenser. This situation does exist, but the magnitude of the loss is small, although I can offer no specific figures with regard to this item.

Mr. Ashley has indicated that the figures given in the text do not include the steam required for purging. Reference to Table III under "Auxiliary Horsepower" indicates that approximately 1 pph of steam is required per ton of refrigeration for steam vacuum refrigeration systems, and approximately 0.75 pph of steam is used per ton of refrigeration for steam driven centrifugal systems. No additional steam is required for lithium bromide absorption systems. This was indicated in the article, and no attempt was made to conceal this fact. It was presented in the manner shown primarily because steam driven centrifugal systems conventionally refer to their steam rate by reference only to the steam used for the turbine itself, and without reference to the steam used for the secondary ejectors.

Mr. Ashley's last comment in regard to noise level cannot be answered specifically with regard to the sound power level output on a frequency band basis, or the sound pressure level output on a frequency band basis. The investigation of this information was done under the auspices of a government agency, and the specific data obtained are not presently available for publication. It can be said, however, that the frequency and power level of the noise was such as to be easily attenuated, and our own experience indicates that the noise of the falling water in the evaporative

condenser is about the same as the ejector noise itself.

The writer sincerely believes that Mr. Ashley's comments have been of assistance in clarifying some of the points which were not too clearly explained in the article, and serve to emphasize some of the characteristics of absorption refrigeration which were neglected, by omission, in the article itself. I sincerely appreciate the opportunity to reply to Mr. Ashley's comments, and hope that the information set forth above will aid in a better understanding of this very interesting refrigeration system.

1752

No. 1752

Thermodynamic Investigation of a Refrigerant Expansion Engine

THOMAS M. OLCOTT HARALD A. BLUM

The principle of substituting a refrigerant expansion engine for the expansion valve of a vapor compression cycle system is well known and correspondingly documented. Published performance data for such an engine, however, are conspicuously lacking.

This investigation consisted of (1) construction of a simple engine, (2) testing the engine and (3) comparison of its performance with the isentropic expansion.

With this device, the recovery of available energy was only as high as 10%. The present analysis shows that the pressure drop and leakage in the valves were major causes of the low energy recovery. If these were eliminated, the percentage recovery could have reached 40%.

Thomas M. Olcott is a graduate student at Southern Methodist University and a Propulsion Engineer with Convair; Harold A. Blum is Professor, Mechanical Engineering Dept., Southern Methodist University. This paper was prepared for presentation at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961.

A theoretical comparison was made between the single circuit refrigeration system, utilizing the expansion engine and the cascade system.

THERMODYNAMIC BACKGROUND

An ideal vapor compression cycle refrigeration system departs from a Carnot refrigeration cycle in three ways. This paper discusses these differences and presents a device that will allow the ideal vapor compression cycle to partially recover some of its losses.

Both the ideal vapor compression cycle and the reversed Carnot cycle are shown on a temperature-entropy diagram (Fig. 1) and on a pressure-enthalpy diagram (Fig. 2). The ideal vapor compression cycle is shown by the points 1, 2, 3, 4, d, 1. It is a constant enthalpy expansion from points 1 to 2, followed by a constant pressure and temperature vaporization to point 3,

then an isentropic compression to point 4, and finally a constant pressure cooling and condensation back to point 1. The reversed Carnot cycle follows the path shown by points 1, b, 3, c, 1. This cycle consists of an isentropic expansion and compression from points 1 to b and from points 3 to c. It also includes a constant temperature condensation and vaporization from points c to d and from points b to 3 respectively.

These two cycles differ in the following manner. First, the vapor compression cycle system includes the area d, c, 4, d, known as the superheat horn. The inclusion of this area causes an increase in both the heat rejected in the condenser and the work required for compression. The vapor compression cycle also fails to recover work in the expansion from points 1 to 2. This is represented by the area b, c, f, 2, b, which is equal to the enthalpy change from point 2 to point b and is therefore equal to the work lost for an adiabatic process. Finally, the vapor cycle suffers a loss of cooling due to the increase in entropy during the throttling expansion. This loss is also shown by the area b, e, f, 2, b on the temperature-entropy diagram.

There are two possible ways that the superheat horn can be removed. To have a constant temperature cooling of the superheated vapor from points c to d would be one method. This would be impossible, since it requires a rise in pressure and a decrease in volume with no input of work. Another way to remove the superheat horn might be to begin the isentropic compression at point g. This, however, requires compression through the wet region.

The remaining losses are those associated with the throttling expansion. These losses can be partially eliminated by allowing the refrigerant to expand against a piston, so that work may be removed from the expansion. If the output of this piston were usefully utilized, there would be both an increase in the net refrigeration effect and a reduction in the work that would have to be supplied. This cycle is represented by the points 1, b, 3, 4, d, 1 on Figs. 1 and 2.

The performance of refrigeration cycles is studied by comparing

Fig. 1 Temperature-entropy diagram

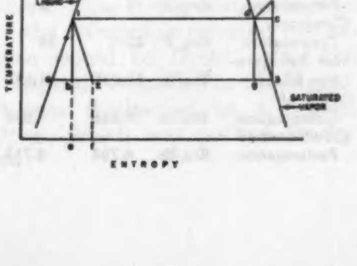
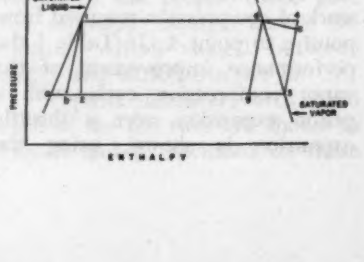


Fig. 2 Pressure-enthalpy diagram



their coefficients of performance. The coefficient of performance (C.O.P.) is defined as the net refrigeration effect divided by the net work that must be performed on the cycle. The net refrigeration effect is equal to the enthalpy change during the constant pressure vaporization. The net work performed on the cycle is equal to the enthalpy change during the compression process minus any work that is produced by any other part of the cycle.

The coefficients of performance of the various cycles discussed are as follows: (See Fig 2)

The ideal vapor cycle with throttle expansion

$$\text{C.O.P.} = \frac{H_1 - H_3}{H_4 - H_1}$$

The reversed Carnot cycle

$$\text{C.O.P.} = \frac{H_1 - H_b}{(H_c - H_1) - (H_3 - H_b)}$$

The ideal vapor cycle with piston expansion

$$\text{C.O.P.} = \frac{(H_1 - H_3) + (H_3 - H_b)}{(H_4 - H_1) - (H_3 - H_b)}$$

$$= \frac{H_1 - H_b}{(H_4 - H_1) - (H_3 - H_b)}$$

It is easy to see that the vapor compression cycle with piston expansion is different from the Carnot cycle only by the additional work of compression required from point c to point 4. In Table I the performance improvement of the vapor compression cycle with a piston expansion over a throttle expansion is shown, using the standard ton characteristics and with Refrigerant 12.

EXPERIMENTAL EQUIPMENT

As the building of an isentropic expansion device would be impossible, it was decided to produce a simple expansion engine and to determine how much of the available work from an isentropic expansion could be recovered.

The engine used was a small single cylinder refrigeration compressor. The valve plate was removed, and the inlet and discharge manifoldings were reamed out to reduce pressure drop. Also the passage leading from the crankcase to the inlet manifold was sealed, and provisions were made to vent the crankcase to a low pressure drum. Electrically operated solenoid valves were placed at the inlet and discharge of the compressor. A wooden fly wheel (eight in. diam) was fastened to the compressor shaft. Strips of linoleum were fastened to the outer edge of the fly wheel, which activated micro-switches, which in turn operated the solenoid valves.

By changing the length of

Table I Standard Ton of Refrigerant 12

		Throttle Expansion	Piston Expansion
Evaporator Temperature	deg F	5	5
Condenser Temperature	deg F	86	86
Net Refrigeration Effect	Btu/lb	50.035	51.633
Work of Compression	Btu/lb	10.636	9.038
Coefficient of Performance	Btu/lb	4.704	5.713

these strips, both the inlet and discharge valves' timing could be varied. Also fastened to the wooden fly wheel was the reducing motion for the engine indicator. This consisted of a slide and crank mechanism. The ratio of the crank length to the eccentric was the same as that of the engine connecting rod length to one half of its stroke. Therefore, the indicator motion was directly proportional to the piston travel at every point. The engine indicator used was a standard Crosby Steam Engine Indicator. Figs. 3, 4 and 5 show a top, front, and back view of the engine.

The engine cylinder and crankcase, as well as the inlet and discharge lines, were well insulated. Both the inlet and the discharge lines were instrumented with pressure gauges. The crankcase of the engine was vented to the discharge line to allow any refrigerant that might leak past the piston to flow to the low pressure drum. This prevented any pressure build up in the crankcase.

The engine was operated by flowing high pressure liquid Refrigerant 12 through it and then allowing the refrigerant to expand to a low pressure source. The high pressure liquid refrigerant was kept in a drum at approximately room temperature. This drum was inverted and fastened to the inlet valve. Thus it was insured that all of the refrigerant entering the engine would be liquid. After the refrigerant expanded in the engine, it was discharged to a second drum, which was packed in dry ice.

Mass flow of the refrigerant was determined by weighing the drums and observing the time of the run. Indicator cards were taken every 30 sec and engine speed was taken continuously by a direct count. This was possible since the engine speed was always about 60 rpm. The pressure in the supply drum, as well as the discharge drum, also was recorded during the run. Approximately 10 lb of refrigerant were used for each of the runs. This resulted in a run of approximately four min in length. Fig. 6 shows the complete test setup.

RESULTS

Ten satisfactory data runs were made during the testing of the engine. The data from these runs are presented in Table II.

The three inlet valve angles that were used were 11.4, 13.9, and 17.6 past top dead center. The supply pressure varied between 80 and 142 psia, and the discharge pressure varied between 25 and 33 psia. The mass flow of refrigerant through the engine was nearly constant, varying between 2.1 and 2.4 lb per min. The speed of the engine varied between 45 and 71 rpm. The indicated horsepower taken from the indicator cards varied between 3.39×10^{-3} and 5.74×10^{-3}. This, divided by the mass flow and converted to the proper units, gave the enthalpy change across the engine, which varied between 0.0621 and 0.0980 Btu per lb.

Typical indicator cards from each of the ten runs are shown in Figs. 7 and 8. Included on these

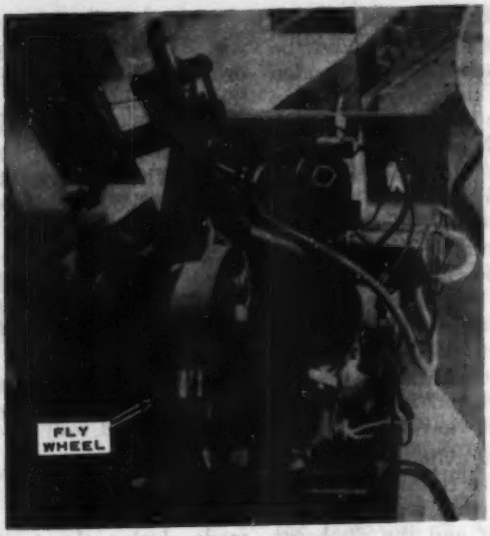

FLY
WHEEL

Fig. 3

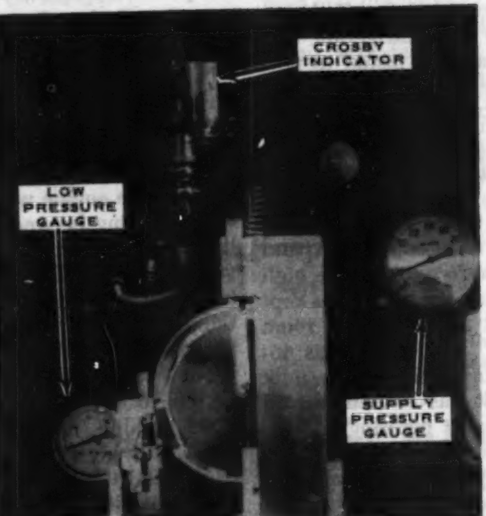

CROSBY
INDICATOR
LOW
PRESSURE
GAUGE
SUPPLY
PRESSURE
GAUGE

Fig. 4

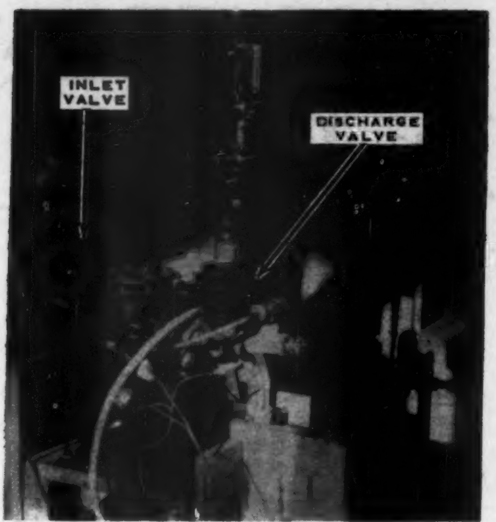


Fig. 5

Fig. 6

Figs. 3, 4, 5, 6 Top, front and rear views of the engine and of the entire test facility

Table II Test Data

Run Number		1	2	3	4	5	6	7	8	9	10
P_1 = Supply Pressure	psia	90	80	80	90	87	142	95	105	86	121
P_2 = Max. Press in Engine	psia	49.6	46.0	48.4	51.9	51.1	56.4	45.0	47.6	42.6	55.5
P^3 = Discharge Pressure	psia	26.5	29.6	28.6	33.1	32.6	26.6	30.2	32.0	25.1	29.3
P_4 = Press. in Low Temp. Drum	psia	15	20	20	25	25	15	20	23	15	20
Mean Effective Pressure	psi	15.8	11.4	12.7	11.5	13.2	21.6	10.8	12.3	12.8	19.4
Indicated Horsepower	hp	0.00413	0.00353	0.00339	0.00368	0.00382	0.00574	0.00444	0.00481	0.00372	0.00530
Indicated Power	Btu/Sec	0.00294	0.00251	0.00240	0.00261	0.00270	0.00405	0.00314	0.00340	0.00264	0.00376
Enthalpy Change Across Engine	Btu/lb	0.0725	0.0632	0.0621	0.0742	0.0702	0.0980	0.0770	0.0865	0.0760	0.0950
Mass Flow	lb/min	2.27	2.38	2.32	2.11	2.31	2.44	2.45	2.36	2.08	2.37
Engine Speed	rpm	45.0	53.0	46.0	55.0	50.0	45.5	71.0	68.0	50.0	47.0
Inlet Valve Angle	deg	13.9	17.6	17.6	17.6	13.9	13.9	11.4	11.4	11.4	11.4
ΔH Available, expanding isentropically from point 1	Btu/lb	1.28	0.78	0.71	0.69	0.83	2.42	1.16	1.21	1.22	1.79
Engine Efficiency, including all losses	%	5.7	8.1	8.8	10.7	8.5	4.1	6.7	7.2	6.2	5.3
ΔH of Engine, eliminating valve losses	Btu/lb	0.179	0.149	0.139	0.142	0.156	0.364	0.222	0.277	0.215	0.297
ΔH available with an isentropic expansion eliminating valve losses	Btu/lb	0.70	0.51	0.43	0.49	0.54	1.42	0.56	0.78	0.73	1.18
Engine Efficiency, eliminating valve losses	%	25.5	29.3	32.3	29.0	29.0	25.7	39.9	35.4	29.5	25.0

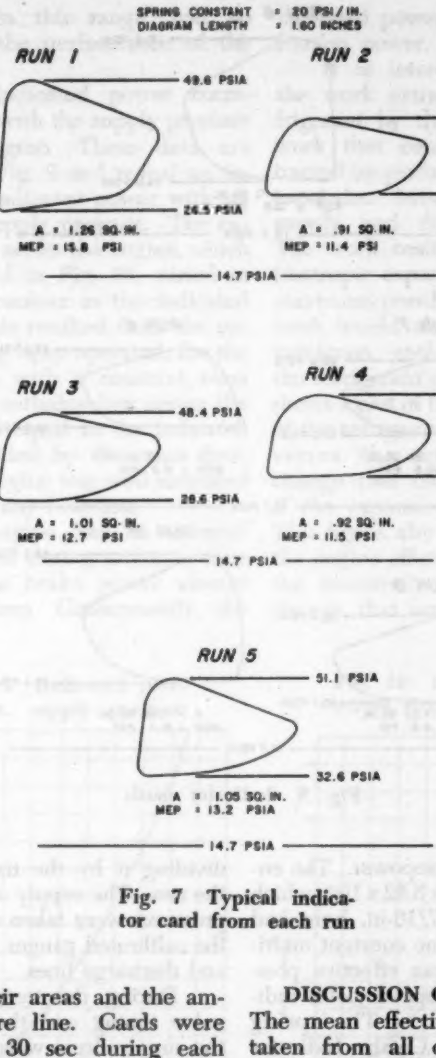


Fig. 7 Typical indicator card from each run

cards are their areas and the ambient pressure line. Cards were taken every 30 sec during each run, which resulted in 5 to 8 cards for each run. The deviation in area from card to card for a given run was never more than 20%.

DISCUSSION OF RESULTS

The mean effective pressure was taken from all of the indicator cards during each of the ten runs. It was then averaged and combined with the average engine speed for that run to determine

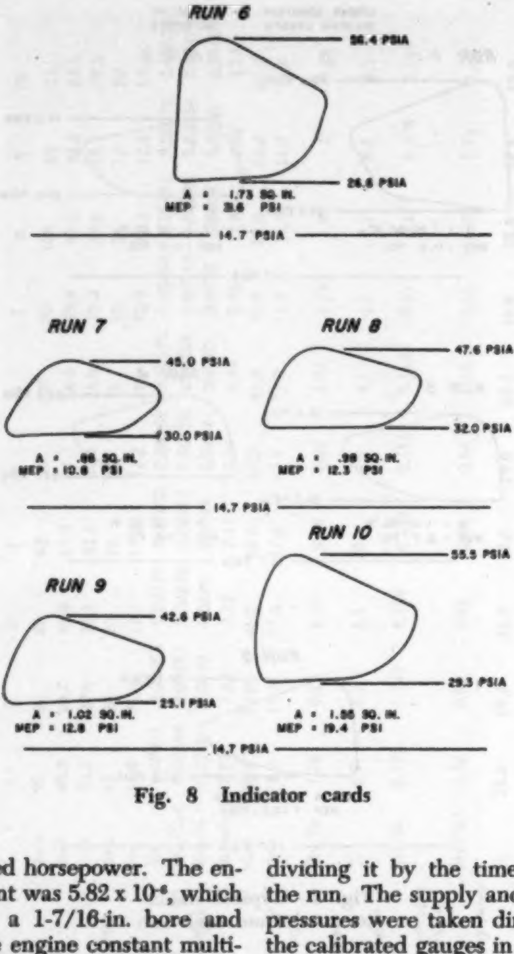


Fig. 8 Indicator cards

the indicated horsepower. The engine constant was 5.82×10^{-6}, which came from a 1-7/16-in. bore and stroke. The engine constant multiplied by the mean effective pressure and engine speed yields indicated horsepower. The spring constant for the Crosby Indicator was 20 psi per in. for all runs. The mass flow of refrigerant was determined by taking the loss of weight of the supply drum and dividing it by the time length of the run. The supply and discharge pressures were taken directly from the calibrated gauges in the supply and discharge lines.

During the testing the inlet valve timing and the pressure in the supply drum were varied. Several runs were made at each of the three inlet valve openings. From examination of the data it appeared that the variation in the inlet valve

opening, in this range, had no effect on the performance of the engine.

The indicated power correlated well with the supply pressure to the engine. These data are shown in Fig. 9 and reveal an increase in indicated power with increasing supply pressure. The enthalpy loss across the engine, which was plotted in Fig. 10, varied in the same manner as the indicated power. This resulted from the engine having been operated, for the most part, with a constant mass flow. The enthalpy loss across the engine was equal to the indicated power divided by the mass flow, since the engine was well insulated to prevent any heat loss.

The engine had no external load applied during any run, thus making the brake power always equal to zero. Consequently, the indicated power was equal to the friction power.

It is interesting to compare the work extracted from the refrigerant by this engine with the work that could have been extracted by performing an isentropic expansion between the engine's supply and discharge pressures. The work realized from such an isentropic expansion would be the maximum possible. This maximum work would also be equal to the maximum enthalpy change that the refrigerant could have. Fig. 11 shows a plot of the enthalpy change of the refrigerant across the engine versus the minimum enthalpy change that could have occurred if the expansion were isentropic. This figure also includes a plot of the engine efficiency, or percent of the maximum available enthalpy change, that was recovered by the

Fig. 9 Indicated power vs. supply pressure

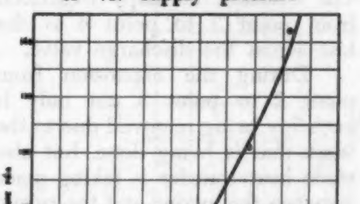


Fig. 10 Enthalpy loss across engine

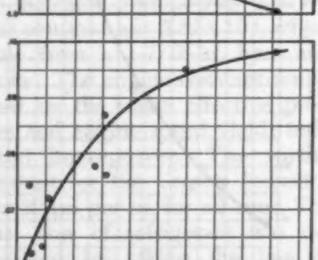
engine. This curve is just the slope of the previous curve. This efficiency varied between 11 and 4%.

FACTORS AFFECTING PERFORMANCE

Reasons for low reported efficiencies can be seen best by studying the process that occurred in the engine during a complete stroke. Fig. 12 is a plot on a pressure-enthalpy diagram showing the conditions of run number 7. Point 1 on this figure is the state of the saturated liquid refrigerant in the supply drum.

As the liquid passed through the inlet valve, there was a considerable pressure loss due to the high flow through the valve. This pressure loss was pure throttling and is shown as a constant enthalpy

Fig. 11 Engine efficiency including all losses



expansion to point 2. The first reason why the efficiency of this engine was so low is this large throttling loss through the inlet valve.

The pressure at point 2 was determined by taking the average maximum pressure from the indicator cards. The measured pressure loss agrees with the calculated value for this size valve and flow conditions. A calculation of this loss for run number 7 is presented in part A of the Appendix. This calculation shows that with a flow of 1.24 lb per sec through the valve there will be a loss of 50 psi across the valve. This agrees with the measured flow of 1.28 lb per sec and the pressure loss of 50 psi.

Point 2 is the maximum pressure in the engine and represents the beginning of the engine expansion. This expansion continues to point 3, which indicates the end of the discharge stroke where the pressure in the engine is the least. The constant enthalpy expansion from point 3 to point 4 is the loss across the discharge valve.

During the expansion from point 2 to point 3 not only is enthalpy being removed due to the work that is being done, but also some heat transfer is taking place between the engine and the refrigerant.

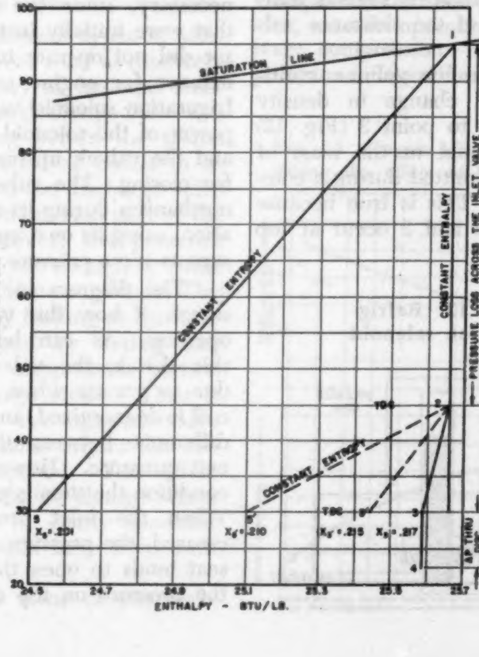
If this test had been performed by perfectly insulating the engine from its surroundings and by then allowing the refrigerant to flow through the engine until equilibrium conditions were reached, the net contribution of any heat transfer effects would have been zero.

Thus, the engine would have maintained an average temperature that would have been between the temperature at point 2 and the temperature at point 3. During the first part of the power stroke some heat would have been transferred from the engine to the refrigerant. As the refrigerant cooled to a temperature below that of the engine, the same quantity of heat would have been returned to the engine. Thus the net enthalpy loss of the refrigerant would have been equal to the work removed.

The reason that this test was not conducted in the manner just described was that the quantity of refrigerant required to cool the engine to equilibrium conditions was a good portion of the amount available for the entire run. Also a continuous method of determining the mass flow of refrigerant would have been required since the testing would have had to begin when equilibrium conditions were reached.

In the preliminary runs while the engine was being prepared, it was determined that the pressure at the beginning of the expansion

Fig. 12 Conditions of expansion for run No. 7

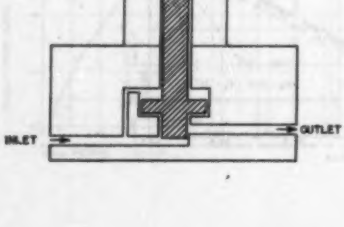


was approximately 45 psia and at the end of the expansion approximately 30 psia with the corresponding saturation temperatures of 30 and 12 F respectively. Therefore, in an attempt to bring the engine to equilibrium conditions before the test began, the engine was insulated with glass wool, and then a few pieces of dry ice were placed near the cylinder to cool it to approximately 15 to 20 F. This temperature was noted by observing the pressure of the saturated gas in the engine crankcase.

The clearance volume of this engine is 6.3 cu in. and the swept volume is 2.3 cu in. This means that the swept volume is only 27% of the total volume. This percentage is quite low, as in most engines the swept volume is several hundred percent of the clearance volume.

The clearance volume multiplied by the change in density from point 2 to point 3 (Fig. 12) should be equal to the mass of refrigerant admitted during a complete stroke. This is true because both points 2 and 3 occur at top dead center, point 3 being just prior to the opening of the inlet valve and point 2 being just after the opening of the inlet valve. For run number 7 the density at point 2 is 6.76 lb per cu ft and at point 3 is 3.58 lb per cu ft. The density difference multiplied by the clearance volume yields a mass of 0.0116 lb. However, the mass admitted during each stroke in run number 7 was 0.035 lb; three times as much as would have been expected.

Fig. 13 Refrigeration solenoid valve



The reason for this huge discrepancy is the other major factor that caused the engine to have such a low efficiency. As was previously stated, the valves used on the engine were electrically operated solenoid valves. These were necessary, since the reed valves that were initially in the compressor did not operate in the proper manner for engine use. The refrigeration solenoid valve uses the power of the solenoid for opening and the valve's upstream pressure for closing. The valve is a servo mechanism during its closing operation, using its own upstream pressure as servo pressure.

The diagram in Fig. 13 is a sketch of how this type of valve operates. As can be seen from this sketch, the valve stem falls due to gravity when the solenoid coil is deenergized, and there is no difference between the inlet and exit pressure. However, in this condition the stem is poorly seated. When the inlet pressure is increased, the pressure on the valve seat tends to open the valve, and the pressure on top of the piston

above the valve seat tends to close the valve. Since the area of the piston is greater than the valve seat, the net effect is to close the valve. This system works quite well for most operations.

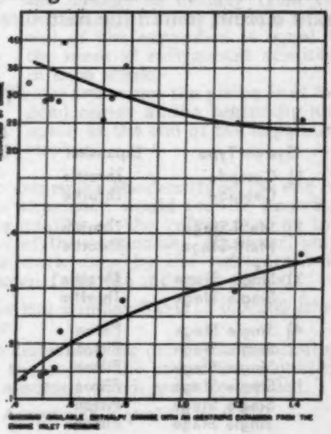
However, when there is a sudden rise in the inlet pressure, a small amount of flow leaks past the valve seat. This is because the flow path to the piston is much smaller than that through the valve. In the case of this engine, the discharge valve leaked nearly two-thirds of the total flow admitted by the inlet valve before it closed tightly. This was observed when the engine was being run on air and was discharging to ambient. When the inlet valve opened, a small burst of air was passed by the discharge valve.

Three different types of solenoid valves were tried as discharge valves, but they all had this tendency. Also the discharge valve was closed 34 deg before the piston returned to top dead center in an attempt to seat the discharge valve during the compression part of the stroke. However, the pressure build up was not enough to cause the valve to close completely. This can be seen by looking at the indicator cards and by noting that there was no compression portion of the discharge stroke. Thus it appears that the exhaust valve closed at the same instant that the inlet valve opened. This effect caused the enthalpy change across the engine to be nearly one-third less than it could have been with better valves.

Of course some of this leakage flow was going past the piston; but since the compressor piston and cylinder walls seem to be in good condition, this leakage was probably quite small. Therefore the two major causes of inefficiency were brought about by the inlet and discharge valves. When these effects are eliminated from the experiment by considering an isentropic expansion from point 2 to point 3 (Fig. 12) using only the mass of refrigerant that remained in the cylinder during the power stroke, the percent of work recovered varies between 25 and 40%. These data are shown in Fig. 14 and the method of calculation is described in part B of the Appendix for run number 7.

Comparison between piston expansion, cascade operation and multistage compression — In low temperature refrigeration systems cas-

Fig. 14 Engine efficiency eliminating the effect of valve losses



cade cycles commonly are used to improve the systems' coefficient of performance. In the cascade system a series of refrigerants with progressively lower boiling points is used in a series of single stage units. The evaporator of the first system, operating at the highest temperature, is used to cool the condenser of the second system; this can be continued for as many cycles as desired. Thus each refrigerant circuit is a system in itself, and each refrigerant can be chosen so that it operates best within the required temperature and pressure range.

Another cycle commonly used to improve C.O.P. is multi-stage compression with both intercooling and subcooling. In this cycle a portion of the liquid leaving the condenser is subcooled by partial vaporization of the remaining liquid. The subcooled portion then passes to the evaporator and on to the first compressor. At the discharge of this compressor the two refrigerant circuits join in the flash intercooler and then pass through the second compressor and on to the condenser.

Table III presents a comparison among the 1) cascade system, 2) multi-stage compressor system, 3) single-stage system with throttle expansion and 4) single-stage system with piston expansion. The load is absorbed at −80 F and rejected at 80 F in each system. Several different refrigerants also are included in the comparison. It can be seen from this table that the single-stage system with piston expansion has the best performance when the expansion efficiency is as high as 75%.

The comparison of these four systems was computed assuming compression to be adiabatic (Appendix, Part C). It should be noted that the corrections for compression efficiency would affect the single stage compression units more unfavorably than the multi-stage compression units.

FUTURE DEVELOPMENT

While the engine described in this

Table III

System Type	Expansion	Expansion Efficiency	Refrigerant	C.O.P.
1) Cascade	Throttle	0%	R 22/R 12	1.76
Cascade	Throttle	0%	R 13/R 22	1.65
2) Multi-Stage	Throttle	0%	R 22	1.92
Multi-Stage	Throttle	0%	R 12	1.88
3) Single Stage	Throttle	0%	R 12	1.69
Single Stage	Throttle	0%	R 22	1.53
4) Single Stage	Piston	50%	R 12	1.79
Single Stage	Piston	75%	R 12	1.88
Single Stage	Piston	100%	R 12	1.97
Single Stage	Piston	50%	R 22	1.80
Single Stage	Piston	75%	R 22	1.97
Single Stage	Piston	100%	R 22	2.15

paper successfully demonstrates the principle of the refrigerant engine, the fraction of the isentropic process available energy actually recovered was too small to be commercially interesting.

To make this system a contribution to the art, this principle should be applied to an operating refrigeration cycle in such a way that work obtained from the engine be used either to reduce the work input to the compressor or handle auxiliary equipment.

APPENDIX

A. The following is an estimate of the pressure drop across the inlet valve to the engine for the conditions of run number 7. This is the pressure drop calculation method outlined by the ASRE DATA BOOK, Refrigeration Applications, 6th Edition, 1949.

Mass flow of refrigerant = 0.0345 lb/rev

Length of time the inlet valve was open = 0.027 sec/rev

Mass flow through the inlet valve = 1.28 lb/sec

Equivalent length of a 3/16-in. orifices solenoid valve = 15 ft of 3/16-in. straight pipe

Flow giving 1-psi pressure drop per 100 linear ft of 2-in. schedule 40 steel pipe = 21,000 lb/hr of Refrigerant 12

Multiplier H_1 to be applied to 2-in. schedule 40 steel pipe to find the flow giving the same pressure drop in 3/16-in. pipe = 0.004

Measured pressure drop across the inlet valve = 50 psi

Multiplier H_2 to be applied to the flow to give a pressure drop of 50 psi = 8.0

Therefore the flow that gives a 50-psi pressure drop in a 3/16-in. orifice solenoid valve is

$$\frac{(21{,}000)\ (0.004)\ (8)\ (100)}{(3600)\ (15)} = 1.24 \text{ lb/sec}$$

B. The following calculation is a determination of the efficiency of the engine if the effect of the inlet valves' pressure loss and the discharge valve's leakage were eliminated. This calculation is done for run number 7

The supply pressure = 95 psia

The enthalpy of the saturated liquid entering the inlet valve = 25.667 Btu/lb

Therefore the enthalpy leaving the inlet valve = 25.667 Btu/lb

The pressure leaving the inlet valve = 45.0 psia

This yields a quality at the beginning of the expansion = 0.156

The density of the refrigerant at the beginning of the expansion = 6.76 lb/ft^3

The entropy of the refrigerant at the beginning of the expansion = 0.0546 Btu/lb F

The pressure at the end of the expansion = 30.2 psia

If the expansion were isontropic the quality at the end of the expansion = 0.210

This yields an enthalpy of 25.109 Btu/lb

Therefore the enthalpy change with an isontropic expansion = 25.667 — 25.109 = 0.553 Btu/lb

The clearance volume of the engine = 0.00365 ft^3

The clearance volume multiplied by the change in density from the beginning of the expansion to the end of the expansion is equal to the mass of refrigerant admitted in each stroke

This is true since the piston is at top dead center at the beginning and again at the end of the expansion stroke.

To determine the density at the end of the expansion would require knowing the quality of the refrigerant at the end of the expansion. However, only the work done by the refrigerant is known from the indicator cards.

For run number 7 this = 0.00265 Btu/rev

To determine the density and enthalpy at the end of the expansion a quality is assumed which must be verified.

Assume this quality = 0.210

Density at the end of the expansion $= 3.57\ lb/ft^3$

The density difference multiplied by the clearance volume yields a mass of refrigerant = 0.0116 lb/rev

This mass divided into the work done by the refrigerant yields an enthalpy change = 0.229 Btu/lb

Therefore, the enthalpy at the end of the expansion = 25.438 Btu/lb

However, this enthalpy occurs at a quality 0.2145

The density at this quality $= 3.47\ lb/ft^3$

Recalculating the enthalpy change from this density = 0.222 Btu/lb

The enthalpy = 25.445 Btu/lb

This enthalpy occurs at a quality = 0.215

This is very close to value used

$$\text{Thus, the engine efficiency} = \frac{0.222}{0.558} = 39.9\%$$

C. The following is a calculation of the coefficients of performance of (1) a cascade system, (2) a multi-stage compression system, (3) a single-stage system with throttle expansion and (4) a single-stage system with piston expansion. All four systems are designed to supply refrigeration at an evaporator temperature of —80 F and to reject heat at a condenser temperature of 80 F.

In the second cascade system the load at —80 F is absorbed by a Refrigerant 22 unit with an 80 F condenser. The Refrigerant 13 condenser is —10 F and the Refrigerant 22 evaporator is —20 F.

Heat absorbed in the R 13 evaporator = (50.6 — 7.5) = 43.1 Btu/lb R 13

Heat rejected in the R 13 condenser = (61.0 — 7.5) = 53.5 Btu/lb R 13

Heat absorbed in the R 22 evaporator = (102.8 — 34.3) = 68.5 Btu/lb R 22

Work of compression for the R 13 unit = (61.0 — 50.6) = 10.4 Btu/lb R 13

Work of compression for the R 22 unit = (123 — 102.8) = 20.2 Btu/lb R 22

Ratio of the mass flow of R 22 to R 13

$$= \frac{53.5}{68.5} = .78\ \text{lb R 22/lb R 13}$$

$$\text{C. O. P.} = \frac{43.1}{10.44 + (20.2)\ (.78)} = 1.65$$

2. In the first two-stage unit with a subcooler and a flash intercooler the intermediate pressure is 27.4 psia, the evaporator —80 F and the condenser 80 F. Refrigerant 22 is the refrigerant. The liquid refrigerant is subcooled to the saturated intermediate pressure before entering the evaporator and the first stage compressor discharge is desuperheated in the flash intercooler.

Heat absorbed in the evaporator = 95.68 — 6.4) = 89.3 Btu/lb

Mass flow through the evaporator

$$\text{for a 1 ton load} = \frac{200}{89.3} = 2.24\ \text{lb/min}$$

Combining the flash cooler enthalpy balance and subcooler enthalpy balance results in an enthalpy of the refrigerant leaving the subcooler and entering the intercooler of 86.0 Btu/lb. This results in a total mass flow through the high stage compressor of 3.45 lb/min. The work of compression for the low pressure stage unit = (80.5 — 68.5) (3.16) = 37.9 Btu/min The work of compression for the high pressure stage unit = (89.5 — 75.5) (4.85) = 68.0 Btu/min

Thus:

$$\text{C. O. P.} = \frac{200}{106} = 1.88$$

3. For a single stage Refrigerant 12 unit with a —80 F evaporator and an 80 F condenser with throttle expansion

Heat Absorbed in evaporator = (68.47 — 23.37) = 45.10 Btu/lb

Work of compression = (96.01 — 68.47) = 27.54 Btu/lb

$$\text{C. O. P.} = \frac{45.10}{27.54} = 1.63$$

A single stage Refrigerant 22 unit with a —80 F evaporator and 80 F condenser with throtle expansion

Heat absorbed in the evaporator = (95.7 — 34.3) = 61.4 Btu/lb

Work of compression = (136 — 95.7) = 40.3 Btu/lb

$$\text{C. O. P.} = \frac{61.4}{40.3} = 1.53$$

4. A single stage Refrigerant 12 unit with piston expansion in lieu of throttling expansion and a —80 F evaporator and an 80 F condenser.

Quality of refrigerant entering the evaporator

$$= \frac{0.05475 + 0.02086}{0.20229} = 0.373$$

Enthalpy of refrigerant entering the evaporator

$$= -8.345 + (.373)(76.312) = 20.35 \text{ Btu/lb}$$

Work performed by the expansion engine if the engine is 100 per cent efficient

$$= (23.37 - 20.35) = 3.02 \text{ Btu/lb}$$

$$\text{C. O. P.} = \frac{45.10 + 3.02}{27.54 - 3.02} = 1.97$$

Work performed by the expansion engine if the engine is 75 per cent efficient

$$= (0.75)(23.37 - 20.35) = 2.26 \text{ Btu/lb}$$

$$\text{C. O. P.} = \frac{45.10 + 2.26}{27.54 - 2.26} = 1.88$$

Work performed by the expansion engine if the engine is 50 per cent efficient

$$= (0.50)(23.37 - 20.35) = 1.51 \text{ Btu/lb}$$

$$\text{C. O. P.} = \frac{45.10 + 1.51}{27.54 - 1.51} = 1.79$$

A single stage Refrigerant 22 unit with piston expansion, a —80 F evaporator and an 80 F condenser.

Quality of refrigerant entering the evaporator

$$= \frac{0.0708 + 0.0255}{0.280} = 0.344$$

Enthalpy of refrigerant entering the evaporator

$$= -10.22 + (0.344)(105.9) = 26.3 \text{ Btu/lb}$$

Work performed by the engine if the engine is 100 per cent efficient

$$= (34.3 - 26.3) = 8.0 \text{ Btu/lb}$$

$$\text{C. O. P.} = \frac{61.4 + 8.0}{40.3 - 8.0} = 2.15$$

Work performed by the engine if the engine is 75 per cent efficient

$$= (0.75)(34 - 26.3) = 6.0 \text{ Btu/lb}$$

$$\text{C. O. P.} = \frac{61.4 + 6.0}{40.3 - 6.0} = 1.97$$

Work performed by the engine if the engine is 50 per cent efficient

$$= (0.50)(34.3 - 26.3) = 4.0 \text{ Btu/lb}$$

$$\text{C. O. P.} = \frac{61.4 + 4.0}{40.3 - 4.0} = 1.80$$

BIBLIOGRAPHY

ARTICLES

Threlkeld, J. L. Nov. 1953. A Thermodynamic Study of Vapor Compression Heat Pump Cycles. Refrigerating Engineering: pg. 1202.

BOOKS

Baker, T. C. 1929. Indicating, Valve Setting, Calculations. International Textbook Co.

Jakob, Max. 1949. Heat Transfer. Vol. 1. John Wiley & Sons.

Jordan, R. C., Priester, G. B. 1956. Refrigeration and Air Conditioning. Prentice-Hall Inc.

Lee, J. F., Sears, F. W. 1953. Thermodynamics. Addison Wesley Publishing Inc.

Porter, A. W. 1931. Thermodynamics. John Wiley & Sons.

Moyer, S. A. 1934. Power Plant Testing. McGraw Hill Inc.

Priester, G. B. (ed.), 1949. ASRE DATA BOOK, 6th Edition.

BULLETINS

Thermodynamic Properties of Freon 12. Bulletin T12: prepared by E. I. duPont de Nemours & Co.

INDEXES, coverage of literature search

Engineering Index. Engineering Index Inc., 1901-1937, 1940-1957, subjects: refrigeration, air conditionng, thermodynamics.

Industrial Arts Index. H. W. Wilson Company, 1914-1956, 1958 Vol. 46 #3, 6, 9, 12, 1959 Vol. 47 #1, 2, 3, 4, subjects: refrigeration, air conditioning, thermodynamics.

DISCUSSION

JOHN C. LEE, Des Plaines, Ill.: The ratio of the compression work to the expansion work depends upon the type of refrigerant that is used. In general, the ratio increases with the molecular weight of the refrigerant simply because the slope of the liquid line is smaller as the molecular weight of the refrigerant is increased. Why does Refrigerant

22 give a higher C.O.P. than Refrigerant 12? Is using the expansion the reason why it is greater than Refrigerant 12?

AUTHOR OLCOTT: Referring to the table, with 100% efficiency, using Refrigerant 22, 2.15 was obtained. It is not certain why, but the difference probably is that if you get inside of the dome as you come down a constant enthalpy line and a constant entropy line, the spread must be further in Refrigerant 22. In this table, it was pointed out that the load absorbed −80 and rejected +80 in every case. So there is a considerable difference between a constant entropic line and a constant enthalphic line from plus A down to the region of minus A.

MR. LEE: Is this because the liquid line is lower in the case of Refrigerant 12?

AUTHOR OLCOTT: That is correct.

1753

No. 1753

Discussion of Some Strength Characteristics of Ice at the Interface

J. K. STENE

W. E. FONTAINE
Member ASHRAE

Several research programs to determine the adhesion strength of ice formed on various surfaces have been reported upon previously. The strength has been calculated based on the average stress required to separate the ice and the surface in either tension or shear.

Sibbitt, Fontaine and Dotson[11] conducted a series of tests to determine relative tensile and shear stress when ice adhered to various metal surfaces. The tests were conducted in a large cold room which was maintained at a temperature of about 20 F. Distilled water was used to eliminate any variables present in tap water. Care was used to apply the load evenly as their preliminary tests showed that the stress was a function of the rate of loading.

In these investigations when ice was formed between two cold metal surfaces, the ice failed at a crystal interface. This was attributed to ice formation advancing on fronts from both surfaces. When one surface was left warm, this type failure was eliminated. However, the breaking plane in the tensile tests always occurred in the ice, but the breaks moved closer to the surface that was originally cold.

Test results in tension were lumped together for all clean metal surfaces. The metals used were brass, copper, aluminum, and steel. The average tensile strength was given as about 105 psi for ice greater than one-eighth in. thick. When tests were performed varying the thickness of the ice, it was

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W. E. Fontaine is Professor of Mechanical Engineering and J. K. Stene is an Associate Professor of Mechanical Engineering at Purdue University. This paper was prepared for presentation at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

found that the stress increased as the thickness decreased.

In another series of tests they treated some metal surfaces with silicone compounds. The average value of the tensile stress for all tests was cited as 8 psi. There was noted a tendency for increasing stress as the number of removals from one surface was increased. This was attributed to removal of the silicone surface with the ice. In this series complete removal of the ice was achieved. In the shear tests the shearing strength of adhesion was determined to be less than the tensile strength of adhesion, averaging about 65 psi, and all separations occurred at the interface and not in the ice.

Results of these tests are illustrated with bar graphs which show the range of stress values for the tension tests and for the shear tests. There is a bar for each class of surface. The frequency of any value is not illustrated, nor is the spread within a range shown.

Jellinek[9] conducted extensive tests on the variation of the tensile stress with variation in geometry. The ice specimens were cylindrical in geometry. The interface was made by freezing the ends of these ice cylinders to stainless steel disks. Water-ice was formed from water that had been prepared by passing through an exchange resin to free it of electrolytes and by boiling to remove air. Snow-ice was formed by sieving snow into water and then freezing the mixture. The metal surfaces used were kept in benzene and cleaned in benzene before the tests.

The tests conducted were at constant temperature and were to measure the tensile strength as a function of the rate of loading, the thickness and the cross-sectional area of the specimens. The results were formulated in terms of the cross-sectional area and the volume. At –4.5 C the following equation applied "over a thousand-fold range of volumes":[9]

$$S = (2.74\,AV^{-0.84} + 9.4)\ \mathrm{kg/cm^2}$$

S = tensile strength

A = cross-sectional area in cm^2

V = volume in cm^3

In the above tests the breaks were partly adhesional (at the interface) and partly cohesional (in the ice). None of the tests reported was for clean separation between the ice and metal. These tests also showed a tendency reported by other investigators; that is, for increasing stress with decreasing thickness. The effect of the rate of loading was that an increase in rate was found to decrease the proportion of the break that was adhesional.

Raraty and Tabor[10] froze ice in a mold so that torque could be applied to a metal cylinder frozen along the axis of the ice cylinder. For the tests torque was applied to the metal cylinder. The mold held the ice stationary so that the failure would occur at the cylindrical interface between the ice and the metal cylinder. Sometimes the failure was in the ice itself.

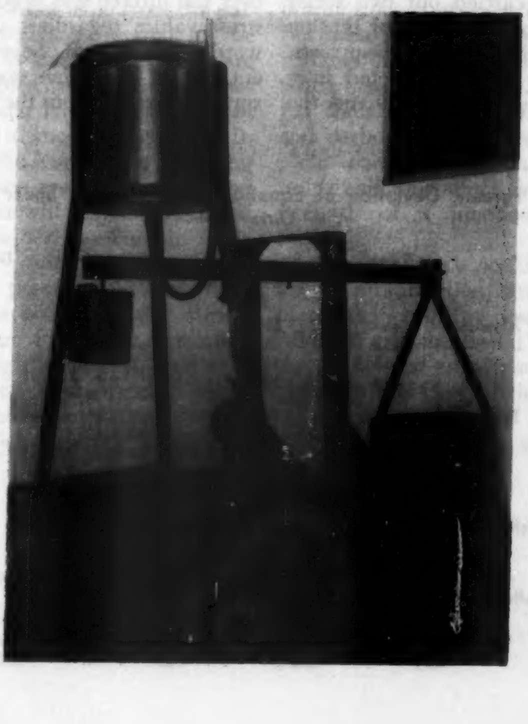
In their experiments the stainless steel surfaces were prepared by polishing, degreasing, cleaning in analar acetone and air drying. At first they boiled the distilled water to liberate the absorbed air,

but when it was noticed that air bubbles tended to be trapped away from the interface this was stopped.

As torque was applied in these torsional shear tests, the investigators found an initial yield was dependent on time. When this variable was eliminated by rapid load application, there were wider variations in the torque values at failure. If the torque was applied slowly and at a uniform rate, there was less scatter in the results. The shear stress at failure was found to be dependent on temperature and increasing almost linearly as the temperature decreased.

Two metals were used in these tests. The rotated cylinder was of stainless steel for the tests shown graphically by the authors, and aluminum was said to have been tested. Less stress was said to be required for failure when stainless steel was used than when aluminum was used. The failure was always in the ice when aluminum was used.

Fig. 1 Lever system at the beginning of the test. Ice is frozen to a vertically positioned surface



These investigators ran another series of tests in which non-metallic substances were used for the rotated cylinder. For the materials used it was found that the stress values increased as the temperature decreased in the range from 0 C to approximately −10 C. In the range from −10 to −30 C the stress value remained essentially constant. A plot of shear stress versus temperature showed that each material had a curve different from each of the other materials. Some further tests using metals covered with thin plastics films gave results similar to the solid plastics and not like the solid metal.

Stein[12] carried out two series of tests. One was a set of tension tests; the other a set of shear tests. These tests were conducted in a cold room where the ice and the environment temperatures varied between 18-22 F. Careful control was kept over the water condition and temperature, and cleanliness was watched carefully in regard to the solid surfaces.

Results of the tension tests showed a large spread in the stress values at separation. This spread was especially large where the break was entirely within the ice. For adhesion to metals, some stress values were over 300 psi; whereas, for adhesion to non-metallic surfaces all values were under 150 psi.

The shear tests showed less deviation from an average value, but some slow sliding of the ice over the surface was observed. All stress values regardless of surface were below 150 psi. Several different metals and some non-metallic surfaces were used in the tests. The

Table I Percent Deviation of Equation from Average Shear Stress Data

Surface Material	Temp R	Stress From Equation psi	Avg. Stress minus Calc. Stress psi	Deviation %
Anodized Aluminum	461	350.7	6.0	1.7
	462	333.8	—10.6	— 3.3
	463	317.8	3.6	1.1
	464	302.6	0.9	0.3
Stainless Steel	462	246.8	1.5	0.6
	463	242.7	— 1.4	— 0.5
	464	238.7	— 0.8	— 0.3
	465	234.8	1.0	0.4
	466	231.0	0.3	0.1
White Teflon	461	73.3	1.8	2.4
	462	70.6	— 2.4	— 3.5
	464	65.5	0.8	1.2
Black Teflon	461	74.5	17.9	24.0
	462	62.9	—17.9	—39.8
	463	53.3	1.7	3.1
	464	45.0	4.4	8.9

metals showed the greater strength of adhesion. Complete removal of the ice was achieved with the shear tests. Extensive tables and graphs are presented to show stress values for each surface and each test.

Several general statements have been cited regarding the difficulties involved in conducting adhesion tests. In one article[10] three basic difficulties were noted. The first concerned the necessity for reproducible cleanliness of the surface. The second concerned the difficulty of applying a uniform load at the interface when working with tension tests. The third difficulty concerned the pattern of stresses at the interface. "The greatest uncertainties arise from the pattern of stresses generated at the interface. In many cases these are very complex and discontinuities may produce large stress concentrations of unknown magnitude. In addition, the very act of forming the interface may leave residual stresses that will play a part in the observed adhesive strength. This occurs for example when clean metals are pressed together; there is evidence to show that strong junctions are formed at the interface although in general little normal adhesion is observed. This is because released elastic stresses pull the junctions apart one by one as the load is removed."[10] Other comments pertinent to conducting adhesion tests are summarized below.

Two generalizations are that the area of real contact is important in determining adhesion forces, and that there are no smooth surfaces since all carefully polished surfaces have peaks and depressions which are large compared with molecular dimensions.[4] A conclusion that can be drawn is that the roughness of a surface can cause strong and weak points in the interface due to the adhesion forces of near points.

Contamination also is seen to be a real factor in any tests. This is true since contamination reduces the area of real contact. Rough surfaces can be contaminated even when handled carefully. Absorbed gas films, especially water vapor and carbon dioxide, prevent complete contact between the two solids and thus affect the adhesion strength of a junction between them.[4]

The physical position of the ice in an adhesion strength test places the ice in the position of an adhesive and the tested surface in the position of an adherent. Comments on this view of the problem are worth considering, as careful control measures for an experiment

Table II Constants for Each Surface in Equation

$$S = c(1 - dt + et^2 - ft^3)$$

Surface	c	d x 10³	e x 10⁵	f x 10⁴
Anodized Aluminum	368.4	4.95	1.3	0.0
Stainless Steel	255.1	1.66	0.2	0.0
White Teflon	76.1	3.78	0.8	0.0
Black Teflon	88.1	16.86	14.4	8.0

can be deduced from such discussion.

In a discussion by Bikerman[2] on adhesive joints, it is indicated that when an adhesive is applied as a liquid the rate of solidification determines the crystal size and the strength of the solid adhesive. At the same time, the thinner the adhesive the stronger the joint, since there is a greater probability of weak spots in larger volumes. These weak spots make stress distribution in larger volumes less favorable than in small volumes. The greater strength of smaller volumes has been well documented in the work of Jellinek[9] and was well known to other investigators.

Raraty and Tabor[10] write of the stress problem involved when ice freezes at 32 F and then is cooled further. There is a difference in the coefficients of contraction of ice and of the solids to which it is frozen. Metals have smaller coefficients than ice so that tensile stresses are set up in the ice, parallel to the interface. Likewise, compressive stresses are produced in the ice when ice is frozen to plastics which have greater coefficients of contraction than the ice.

Ingersoll, et al,[5] discuss the orientation of crystals when water freezes. The orientation being that the long axis is normal to the cold surface to which it is freezing. Truby[13] studied ice with an electron microscope to ascertain a relationship between the structure of ice and the rate of growth of the ice. All his samples indicated a micro structure of coordinated hexagonal prisms in bundles. The size of a bundle varied from one-half to 20 micron in width and from one to ten micron in length. The microscope pictures failed to show any relation between the size distribution and the rate of growth. Observation of the growing surface of the freezing ice showed pyramidal pits with hexagonal bases.

Bjerrum's[3] discussion of ice structure finds ice to be plastic in behavior because of easy slippage along hexagonal layers of the crystal. Nakaya,[8] in extensive tests on the formation of Tyndall flowers (voids in the ice due to internal melting caused by radiation), found that internal melting followed by refreezing left voids in the ice which were filled with water vapor. He determined that milky layers in ice were layers of these

Table III Temperature and Average Shear Stress for Removal of Ice from Various Surfaces

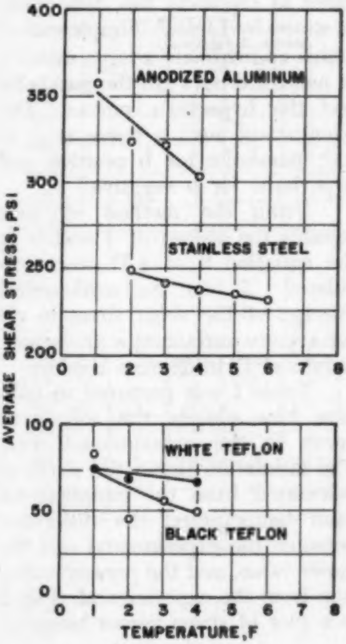
Surface Material	Temp F	Average Shear Stress, psi
Anodized Aluminum	1	356.7
	2	323.2
	3	321.4
	4	303.5
Stainless Steel 1018	2	248.3
	3	241.3
	4	237.9
	5	235.8
	6	231.3
White Teflon	1	75.1
	2	68.2
	4	66.3
Black Teflon	1	92.4
	2	45.0
	3	55.0
	4	49.4

tiny cavities. Under these conditions there are indeterminate internal stresses.

All tests conducted in the strength of adhesion experiments can be summarized qualitatively by saying that some organic surfaces are effective in the release of ice, whereas most inorganic surfaces are ineffective.

In the report of Berghausen, et al,[1] it is stated that "It seems unlikely, ... , that any wholly inorganic coating or material of the type of most metal oxides or salts will ever be found such that the adhesive bond to ice is weaker than the cohesive strength of ice. Very probably the only types of coatings of surface treatments which will lead to ready release of ice from surfaces are those in which a surface of covalently-bound atoms is presented to the ice." This statement covers all results noted above, especially the results that accrued from tension tests.

Fig. 2 Stress versus temperature



EXPERIMENTAL DISCUSSION

The test apparatus and equipment used was that prepared by Stein[12] with some modifications. The shear puller shoulders were widened so that after separation, the corners of the puller were not dragged across the face of the tested surface. The mold shape was changed from square to round, and larger holes for air escape were made in the mold. The can containing shot had a conical bottom placed inside to improve the feed of the shot. Tare weights were made to eliminate one of the weighing steps.

Fig. 1 shows the ice frozen to a vertically positioned surface. The pull was applied in a vertical direction by a simple lever system. With the tare in the weighing bucket and the apparatus in place, the lever arm was balanced. The force necessary for separation was measured as the amount of shot added to the bucket to cause separation. This value was adjusted for the lever arm ratios. The rate of flow of the shot into the bucket was checked to be sure that uniform rates of force application applied to all tests.

Preliminary tests were conducted to establish a uniform

method of preparing and handling the water and for cleaning and preparing the equipment. It was felt that this procedure gave as uniform results as possible.

The most uniform ice formation apparently resulted when the holder and the surface were precooled to the test temperature, the puller and the mold inserted at room temperature and the water poured into the molds at some chilled temperature. Throughout this experiment the water was kept in a refrigerator at 40 F up to the time of pouring. This procedure allowed the ice to freeze with crystals normal to the surface and to reach the coldest temperature in the same time for each test. No imperfections were noticed at the interface when this procedure was followed.

No large change in stress was expected over the range of temperatures reported here. The stress values at higher temperatures were considerably lower, but failure of the interface exhibited definite sliding of the ice over the surface before the sudden release which was looked for as the end of the test. This sliding was visible on the apparatus during the test and also on the test surface after separation.

Failure of the joint with sliding did not agree with the definition of failure followed in this discussion and therefore was eliminated. Failure following sliding indicates that failure is a function of the loading rate. Several checks were made on this point by stopping the loading as soon as sliding was observed. Sliding continued until failure occurred. The total load was less than would have been applied had the load been added continuously during the period while sliding occurred and before the ice left the surface. The sliding did not stop when the loading stopped.

THEORETICAL DEVELOPMENT

When the data from the experiments were plotted on coordinates of average shear stress against temperature for each surface, it was observed that there was a definite tendency for the average shear stress values to decrease as the temperature increased. This suggested the use of a hyperbolic curve to represent the data since, as stated by Lipka,[7] "Simple curves which approximate a large number of empirical data are the parabolic and the hyperbolic curves. The equation of such a curve is $y = a\,x^b$, parabolic for b positive and hyperbolic for b negative."

Using the method of least squares the values of a and b in the equation $S = a\,T^b$ were calculated. S was the arithmetical average of the shear stress in psi for a given surface at a given temperature, T, in degrees Rankine.

Table I was prepared to indicate how closely the calculated curve fit the experimental data. The tabulation shows the stress as calculated from the equation for each temperature, the difference between the experimental and the curve value, and the percent variation from the experimental. Fig. 2 is a plot of stress versus tempera-

ture in degrees Fahrenheit. The curve is the derived equation and the spotters represent the arithmetical averages of the experimental data.

When the absolute temperature was used, the equations involved constants of quite large magnitude so that to obtain more meaningful results further manipulation was used. The Rankine temperature, T, was replaced here by 460 + t, where t is in degrees Fahrenheit. Then the equation $S = a(460 + t)^b$ was expanded by the binomial theorem. In the resulting expansion only those terms were recorded which were needed to keep the shear stress value within plus or minus 0.5 psi of the value calculated with the original equation. The resulting equation is of the form, $S = c(1 - dt + et^2 - ft^3)$. Table II is a tabulation of the values of the constants for each surface which satisfy the above conditions.

DISCUSSION OF DATA AND RESULTS

Table III gives for the four surfaces tested the arithmetical averages at several temperatures in the temperature range, 1-6 F. The results shown in Table III and in Fig. 2 indicate a definite tendency toward decreasing strength with increasing temperature and large differ-

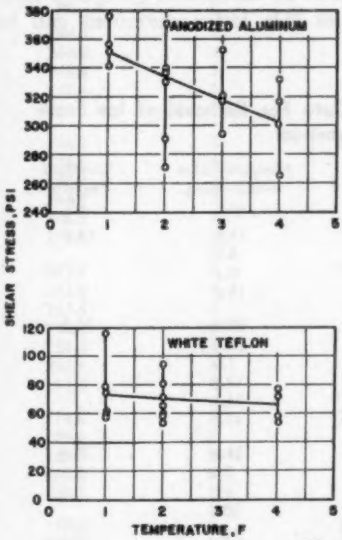


FIG. 3

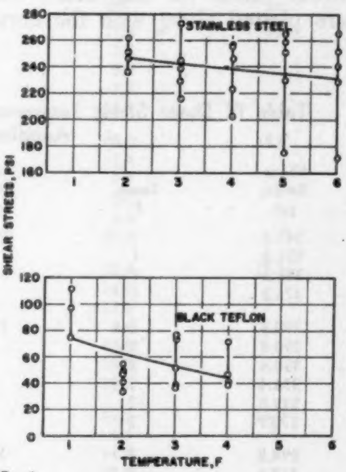


FIG. 4

Figs. 3 and 4 show the shear stress versus temperature for individual tests. Figs. were plotted with curve that fits the desired equation

ences in adhesive strength for the various types of surfaces. The ice adheres most strongly to the anodized aluminum surface, next to the stainless steel, and then much more weakly to the two Teflons, white and black.

Ice will adhere more strongly to an oxidized metal surface; the anodized surface being a completely oxidized surface verifies this statement. According to results mentioned in the literature survey, the data are thus in correct order. The stress for metallic surfaces is greater than for organic surfaces and the stress for the completely oxidized surface is greater than that for the incompletely oxidized surface.

In Figs. 3 and 4 the shear stress values for individual tests are plotted along with the curve that fits the derived equation. These same values are tabulated in Tables IV, V, VI and VII. The values have been grouped numerically from the lowest to the highest for each temperature. These tables have been included as the statistical analysis of the data. The arithmetical mean has been tabulated as, "the arithmetic mean of a sample is the best estimate of the mean of a population."[9] The standard deviation is included as, "the standard deviation of a sample is the best estimate of the scatter in the population and is used to represent the spread."[9] The same author also claims that a sample usually has two-thirds of the data within the spread, arithmetic mean plus or minus the standard deviation. In the fifteen groups of samples this latter statement can be

Table IV Shear Stress Temperature Data For Removal of Ice From Anodized Aluminum

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
342.2	1	356.7	14.5	14.4
351.8	1		4.9	
356.6	1		0.1	
376.3	1		19.6	
270.8	2	323.2	52.4	36.2
290.8	2		32.4	
330.4	2		7.2	
336.4	2		13.2	
339.8	2		16.6	
370.7	2		47.5	
294.8	3	321.4	26.6	23.6
317.6	3		3.8	
320.8	3		0.6	
352.8	3		30.9	
265.3	4	303.5	38.2	28.6
300.2	4		3.3	
316.5	4		13.0	
332.1	4		28.6	

seen to be true.

From Table I, it can be seen that the deviations are small except for the black Teflon. This result was expected as the black Teflon surfaces used were not uniformly alike in roughness as were the other surfaces. Thus, these results have an additional variable involved.

Numerical accuracy of the stress values is high. The one measurement involved was the gross load of the load pan and lead shot at the instant of separation of the interface. The scale used was checked for accuracy with dead weights and was found to be accurate to the ounce. At the same time the sensitivity was determined and even at the loads involved the scale was found to be sensitive to weights of less than one ounce. These loads were multiplied by three, the lever arm ratio, which was correct as measured in several trials made with an engraved, steel scale. The ounces were converted to the nearest one-tenth of a pound. The average values then should be correct to within two-tenths of a pound.

Table V Shear Stress Temperature Data for Removal of Ice from Stainless Steel

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
235.5	2	248.3	12.8	9.4
246.0	2		2.3	
248.6	2		0.3	
249.6	2		1.3	
261.8	2		13.5	
216.2	3	241.2	25.0	19.0
228.8	3		12.4	
239.3	3		1.9	
244.9	3		3.7	
245.6	3		4.4	
272.6	3		31.4	
202.5	4	237.9	35.4	21.5
222.9	4		15.0	
224.4	4		13.5	
246.2	4		8.3	
255.4	4		17.5	
257.1	4		19.2	
257.4	4		19.5	
174.6	5	235.8	61.2	36.8
229.6	5		6.2	
250.5	5		14.7	
258.8	5		23.0	
265.7	5		29.9	
170.8	6	231.3	60.5	36.4
228.6	6		2.7	
240.0	6		8.7	
251.4	6		20.1	
265.5	6		34.2	

Table VI Shear Stress Temperature Data for Removal of Ice from White Teflon

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
57.6	1	75.1	17.5	24.2
60.0	1		15.1	
61.9	1		13.2	
75.0	1		0.1	
78.2	1		3.1	
117.8	1		42.7	
53.1	2	68.2	15.1	13.6
58.4	2		9.8	
59.3	2		8.9	
61.7	2		6.5	
66.0	2		2.2	
72.2	2		4.0	
81.0	2		12.8	
93.8	2		25.6	
54.4	4	66.3	11.9	10.9
56.4	4		9.9	
58.9	4		7.4	
72.6	4		6.3	
77.1	4		10.8	
78.2	4		11.9	

Table VII Shear Stress Temperature Data for Removal of Ice from Black Teflon

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
74.8	1	92.4	17.6	15.8
96.9	1		4.5	
105.4	1		13.0	
32.8	2	45.0	12.2	8.3
40.9	2		4.1	
47.1	2		2.1	
48.8	2		3.8	
54.4	2		9.4	
36.0	3	55.0	19.0	18.9
38.4	3		16.6	
51.6	3		3.4	
72.6	3		17.6	
76.5	3		21.5	
38.4	4	49.4	11.0	15.3
41.3	4		8.1	
46.1	4		3.3	
71.8	4		22.4	

CONCLUSIONS

Work carried out in this experiment, including the preliminary phases, confirms that ice is not a uniform structure of crystals that has a unique strength of adhesion. Rather, ice is formed with imperfections between crystals that make weak spots, which influence the results obtained from strength tests. However, on the average there is a strength of adhesion which is a characteristic of the material to which the ice is frozen and which is a function of the temperature. It is believed that tests carried out in the same manner will provide results within the same limits determined in this experiment. It is also believed that the equations derived may be extrapolated beyond the range involved. According to observations in this experiment, if carried above 10 F, slow sliding will be present before the sudden failure. The prediction for the failure strength of thicker samples of ice is also open to question, as the larger volumes provide greater possibility of flaws in the ice which will produce different results.

REFERENCES

1. Berghausen, P. E., and R. J. Good, G. Kraus, B. Posolsky, W. Soller: Fundamental studies of the adhesion of ice to solids. WADC Technical Report 55-44, (1955).
2. Bikerman, J. J.: Surface Chemistry Theory and Applications. Academic Press Inc., New York (1958) (2nd).
3. Bjerrum, Niels: Structure and properties of ice. Science, vol. 25, pp. 385 (1952).
4. Bowden, F. P. and D. Tabor: The adhesion of solids. Structure and Properties of Solid Surfaces, Chicago Press (1953).
5. Ingersoll, L. R., O. J. Zobel, and A. C. Ingersoll: Heat Conduction with Engineering, Geological and Other Applications. Univ. of Wisconsin Press, Madison (1954) Revised.
6. Jellinek, H. H. G.: Tensile strength properties of ice adhering to stainless steel. SIPRE Research Report 23, (1957).
7. Lipka, Joseph: Graphical and Mechanical Computation. John Wiley and Sons, New York (1918).
8. Nakaya, Ukichiro: Properties of single crystals of ice, revealed by internal melting. SIPRE Research Paper No. 13, (April 1956).
9. Quenouille, M. A.: Introductory Statistics. Pergamon Press Ltd., London (1950).
10. Raraty, L. E. and D. Tabor: The adhesive and strength properties of ice. Proceedings of the Royal Society, vol. 245, pp. 184-201 (January 1958).
11. Sibbitt, W. L., W. E. Fontaine, and J. P. Dotson: Ice formation on metal surfaces. Refrigerating Engineering, vol. 62 no. 12, pp. 49-51, 92, 94 (1954).
12. Stein, Ray J.: The Removal of Ice from Metal Surfaces. A Master of Science Thesis, Purdue University (1957).
13. Truby, Frank K.: Hexagonal microstructure of ice crystals grown from the melt. Journal of Applied Physics, vol. 26, pp. 1416-1420 (December 1955).

DISCUSSION

RAY C. EDWARDS, Pompton Plains, N. J.: What is the shearing stress of ice itself as compared to the adhesion strength? If there were a hollow of metal at that point in the surface and there was ice there, how would you shear it off?

AUTHOR STENE: The surfaces which we were treating, with the possible exception of the teflon surfaces which do have some ability to bend, were really flat surfaces. They were checked by various methods for being flat as well as smooth. The condition you mention was not investigated.

R. C. EDWARDS: It was stated that some ice stuck to the surface itself. My question is were you, in some cases, measuring the shearing strength of the actual ice itself or the adhesive strength?

AUTHOR STENE: The case where the ice actually stuck to the surface was that of pulling the ice from the surface. In this case, it is quite certain that a combination of shearing strength in the ice as well as adhesion strength on the surface is obtained. In the set-up for the tests which were conducted and which are shown here, the shearing strength of the ice itself was not obtained because in these tests there was no ice left on the surface; all the ice was completely removed from the surface.

L. F. ALBRIGHT: It is interesting to note that anodized aluminum was used. Would there have been any difference if plain aluminum had been used? In other words, you are probably getting some surface phenomena other than with just the plain aluminum metal.

AUTHOR STENE: A few tests in the preliminary investigations were made with plain aluminum. However, with the plain aluminum, you run into the difficulty of not being able to use the same surface from the point of view that aluminum oxidizes rather rapidly. If one sample is tested after it has been polished after an hour and another is tested twenty-four hours after polishing, you are not getting the same surface. This is one reason for using the anodized surface which is a completely oxidized surface and which would remain relatively clear. The samples were not polished after they were anodized.

L. F. ALBRIGHT: With the oxygen on the surface, some sort of hydrogen bonding might occur.

AUTHOR STENE: This is correct.

L. F. ALBRIGHT: It was mentioned that least squares was used. Were you using a straight line, because you would normally assume a certain form of equation, then apply the least square technique?

AUTHOR STENE: A straight line was not used. An equation of the form $S = aT^b$ was assumed and the method of least squares was used to determine the values for a and b.

C. M. ASHLEY, Syracuse, N. Y.: Were any tests run with a sufficient amount of salt or material of that character so that those would trap dry? In other words, was salt used in the water so that there could be concentrations of brine present. There would have to be a fair amount otherwise it would simply freeze out. That is, it would separate out in the freezing process. But if it is used sufficiently, it will trap. Did your tests include a mixture of this sort?

AUTHOR STENE: None of these tests included any brine in them. The brine would, of course, introduce another problem into the structure of the ice.

1754

No. 1754

Heat and Mass Transfer in Dehumidifying Surface Coils

W. L. BRYAN
Member ASHRAE

In an air conditioning heat exchanger, such as a cooling coil, both heat and water are removed from the air. The added complication of vapor diffusion together with the transfer of heat confronts the refrigeration and air-conditioning engineer with special problems. The engineer has been faced with finding a proper method for rating and making performance calculations for the heat and mass transferred. Much research and analysis has been devoted to this problem and yet the fundamental basis for the methods now in use is debatable. In an attempt to achieve simplicity in application, assumptions have been made or empirical equations proposed. Aside from the loss in accuracy, especially if the operation conditions differ from the conventional range, the basis for the assumptions has not been fully justified.

Each of the commonly used methods, Contact Mixture Analogy, Carrier,[1] and Humidity Method, Tuve and Seigel,[2] assumes that the "Lewis relation," Lewis,[3] applies. Lewis stated that the coefficient of heat transfer divided by the coefficient of vapor diffusion through a gas film is constant and equal to the humid specific heat of the gas. This relation was derived for the case of unsaturated air flowing over a free water surface and does apply with accuracy in the case of proper wet-bulb determination as shown by Carrier[4] and others,[5] but the assumption that this relation applies to dehumidifying coils has not proven correct.[6] One is justified in assuming a Lewis relation at one only on the basis of simplicity with an accepted loss in accuracy. The relatively more complicated enthalpy potential methods[7,8] also are

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based on a Lewis relation equal to one.

Based on the assumption that the Lewis relation applies, the surface temperature of the coil for heat transfer is obtained by calculation. Although this temperature may be specified as a hypothetical value, it must be considered an approximation to the actual surface temperature required in determining the temperature of the refrigerant inside the coil. For a deep, extended surface coil or one operating at moderate load ratios, the accuracy is usually sufficient for engineering purposes. In the case of bare tube coils or shallow coils, the actual surface temperature may be greatly different from the calculated value.

In this first report on the cooperative research project, experimental results are given for the transfer of mass and heat by a bare surface coil under conditions of measured surface temperature. Experimental data are given for the heat and mass transfer coefficients and a simple fundamental equation is given for the actual coil surface temperature.

EXPERIMENTAL APPARATUS

A line diagram of the experimental apparatus is shown in Fig. 1. The apparatus was located on the balcony of the Mechanical Engineering Laboratory, which was large and well ventilated, so that recirculated laboratory air could be used. The intake air was humidified and heated before the fan and metered after a duct run. The sensible and latent condition of the air was determined before and after the coil by wet-bulb and dry-bulb thermometers and dry thermocouples. Care was taken to mix the air after the coil and check the leaving air condition after a set of straighteners. The air was metered again before discharge to the room.

Considerable work was done on the calibration of the temperature and flow measuring equipment. Orifices and nozzles used to meter the flow were checked by combination and pitot tube traverses and all flow pressure measurements were made with micromanometers. Matched mercury in glass thermometers of 0.1 F accuracy and copper-constantan thermocouples through a type K potentiometer with an electronic null indicator were used to measure temperature.

The six experimental coils in

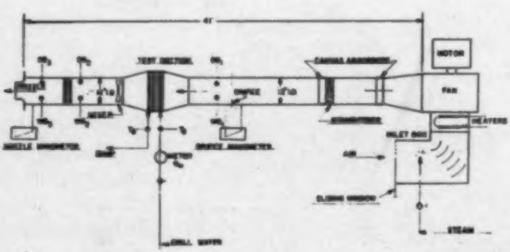


Fig. 1 Diagram of the experimental apparatus

staggered arrangement formed the test section, giving a six-row deep staggered bare coil, Fig. 2.* Each coil was insulated electrically and supplied through rubber connectors with cooling water from a supply header and return header such that the water flowed in parallel through each coil. With a high rate of water flow, the refrigerant temperature could be maintained substantially constant throughout the coils.

The actual wall temperature of the individual tube coils was measured by using each coil as a resistance thermometer in a Kelvin bridge circuit. The bridge circuit read the tube resistance to ½ micro-ohm for a resistance-temperature ratio of 4 micro-ohm per degree F for a tube surface temperature accuracy of better than 0.2 F. The temperature of any one of the six rows was determined independently of the others.

The heat and mass transferred were determined by the temperature and humidity measurements of the air before and after the coil. The rate of mass transfer was checked also by measuring the rate of condensate from the coil.

RESULTS

Tabulated data and results of the experimental tests on the six-row bare coil are given in Table I. The surface temperature of each coil row was determined separately; however, the variation from row to row was usually less than ½ F so an average surface temperature for the total coil is given in the table. The sensible heat transferred and the latent heat, or mass transfer, tabulated are mean experimental values calculated from the data by enthalpy balance, temperature change and condensate measurements.

Since the refrigerant and thus the tube surface temperatures were substantially constant throughout the coil, the heat transfer coefficient was determined by using a logarithmic-mean temperature difference between the entering and leaving air and the mean surface temperature of the coil.

The heat transfer coefficient is defined:

$$dQ_s = h\,(DB - t_s)\,dA \qquad (1)$$

and

$$dQ_s = M_a\, c_p\, dDB \qquad (2)$$

* Data under Fig. 2 for experimental coil:
Tubing 0.500 in. O.D., 0.450 in. I.D., copper

Outside surface area	8.826 ft^2
Face area	1.911 ft^2
Net flow area	1.421 ft^2
Ratio outside to inside surface area	1.111

Fig. 2 Experimental test section for six-row bare coil

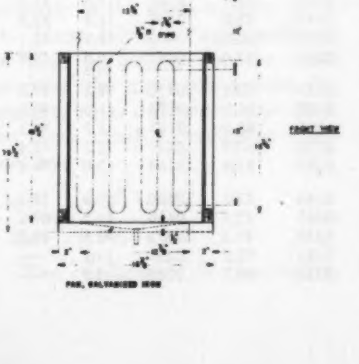


Table I Data and Results

1 Run No.	2 Face Velocity fpm	3 Coil Surface Temp. F	4 Heat Transferred Sensible Btu/hr	5 Latent Btu/hr	6 Entering Air Temp. DB_1	7 Humidity Ratio W_1	8 Leaving Air Temp. DB_2	9 Humidity Ratio W_2	10 Coefficients Heat h	11 Mass h_D
1	127	43.2	2690	2430	99.9	.0185	83.0	.0162	7.2	22.6
2	130	42.0	2620	2690	92.9	.0185	82.5	.0160	6.5	24.8
3	150	44.4	2900	4540	97.2	.0240	87.3	.0206	6.9	28.3
4	152	44.1	2840	4620	96.7	.0242	87.1	.0205	6.7	30.3
5	163	55.2	2370	4930	96.6	.0271	88.2	.0234	7.3	33.3
6	164	55.5	2490	4690	94.8	.0257	87.1	.0224	8.0	32.9
7	164	43.4	3230	5540	99.6	.0260	89.6	.0237	7.2	29.5
8	166	43.3	2130	5620	98.3	.0254	88.7	.0237	7.1	25.9
9	167	42.0	2770	2760	90.2	.0157	81.7	.0141	6.1	24.1
10	172	47.0	2570	1360	88.0	.0130	80.5	.0117	7.7	37.7
11	181	70.0	2620	2270	105.5	.0240	97.9	.0225	9.4	31.3
12	182	42.9	2890	1970	89.0	.0148	80.8	.0135	7.8	25.2
13	187	74.3	2310	1360	104.4	.0229	97.9	.0219	9.8	37.7
14	189	43.2	3390	4790	92.3	.0247	83.0	.0216	8.6	29.7
15	190	42.9	3320	5210	91.5	.0216	82.5	.0183	8.6	37.7
16	193	43.3	3070	4220	90.5	.0198	82.3	.0176	8.1	29.2
17	204	64.1	2440	2910	96.7	.0226	90.6	.0209	9.4	35.2
18	204	64.3	2570	2950	97.4	.0229	91.0	.0209	9.8	28.4
19	206	43.0	2940	2680	107.0	.0141	80.1	.0131	8.2	26.3
20	213	42.2	3130	1880	87.8	.0135	80.3	.0123	8.5	26.9
21	226	42.5	2900	3440	83.5	.0172	77.0	.0154	8.7	34.5
22	235	42.2	3220	1980	87.2	.0180	80.2	.0119	8.8	31.4
23	245	44.6	3300	3490	88.5	.0164	81.6	.0148	9.2	36.0
24	266	42.5	3130	2350	82.0	.0141	76.0	.0130	9.8	32.4
25	267	45.1	3290	3430	88.3	.0159	82.0	.0147	9.4	35.3
26	268	42.8	3400	2820	84.4	.0146	77.9	.0133	10.1	36.7
27	273	46.9	4220	6400	97.9	.0254	90.0	.0225	10.2	42.2
28	274	75.5	4400	6620	98.6	.0254	90.4	.0227	10.7	38.6
29	275	56.4	3810	5750	100.0	.0257	92.9	.0233	10.8	40.1
30	280	42.9	3090	3550	80.7	.0158	75.1	.0146	10.0	33.4
31	284	73.0	2830	3740	103.1	.0278	97.9	.0262	11.8	44.0
32	285	73.0	2800	3680	103.0	.0278	97.9	.0265	11.5	33.1
33	290	48.3	4690	2830	96.2	.0147	88.0	.0135	11.8	43.3
34	292	66.4	3540	4940	106.6	.0272	100.3	—	10.9	43.1
35	302	55.5	4290	6100	104.3	.0253	97.0	.0227	10.8	44.9
36	356	45.3	4440	4960	89.6	.0187	83.1	.0170	12.2	39.0
37	358	46.2	3480	4460	83.2	.0188	78.2	.0173	11.0	42.3
38	358	47.1	3840	5960	86.5	.0207	80.9	.0187	11.9	46.2
39	358	54.0	4070	6030	93.8	.0230	87.9	.0211	12.2	43.4
40	359	45.3	4150	4140	87.6	.0182	81.6	.0168	12.0	46.8
41	361	63.6	2170	1200	86.3	.0164	83.2	.0160	11.6	35.3
42	363	64.6	2040	1510	86.3	.0167	83.4	.0160	11.2	47.2
43	363	73.2	2730	4600	95.2	.0288	91.3	.0272	12.5	50.2
44	369	85.5	1470	—	101.3	.0282	99.2	.0281	11.2	—
45	370	85.7	1610	—	102.0	.0262	99.7	.0262	11.9	—

Table I — Continued

46	370	45.2	4050	3550	84.6	.0169	74.0	.0157	12.2	38.0
47	370	47.0	4250	6530	88.1	.0223	82.2	.0204	11.3	40.8
48	370	44.3	4480	6180	88.9	.0233	82.7	.0212	11.9	43.2
49	371	85.4	1540	—	100.3	.0280	98.1	.0257	12.5	—
50	371	85.5	1610	—	100.6	.0265	98.3	.0266	13.1	—
51	374	74.6	2540	4840	101.4	.0300	97.9	.0286	11.5	42.6
52	374	45.6	4350	5550	88.0	.0221	82.0	.0203	12.3	40.2
53	375	74.5	2620	4800	101.5	.0301	97.9	.0286	11.8	46.7
54	376	46.6	4370	4050	89.6	.0176	83.6	.0163	12.4	41.6
55	377	49.1	3890	9320	88.0	.0268	82.8	.0238	11.8	55.7
56	378	47.1	3770	2440	85.0	.0132	80.0	.0124	11.9	42.4
57	378	48.5	4260	6980	88.5	.0232	82.8	.0209	12.5	50.2
58	379	48.1	4620	6020	92.2	.0210	86.0	.0191	12.5	50.2
59	379	48.1	3770	6090	86.0	.0214	81.0	.0195	11.9	50.7
60	381	53.1	4810	7840	103.9	.0272	97.4	.0247	11.4	46.8
61	382	53.3	4600	7380	102.3	.0263	96.1	.0236	11.5	56.0
62	383	63.7	3440	5960	99.3	.0274	94.7	.0251	11.9	56.1
63	385	52.7	4640	7160	101.8	.0270	95.6	.0233	11.3	48.0
64	386	49.4	5490	7260	99.2	.0246	92.0	.0186	12.7	47.7
65	387	48.1	5560	6210	99.9	.0225	92.5	.0207	12.5	41.6
66	389	63.7	3490	5830	98.3	.0265	93.7	.0247	12.2	44.6
67	389	49.6	6250	9080	107.0	.0287	98.7	.0259	12.6	44.3
68	393	46.8	4550	8330	94.0	.0262	88.1	.0236	11.6	49.6
69	406	48.3	4290	6120	93.5	.0278	87.0	.0209	11.6	43.2
70	436	42.2	4230	4190	83.5	.0146	78.6	.0135	12.3	51.9

by combining an integration, the coefficient for the total coil is:

$$\frac{hA}{M_a c_s} = \ln \frac{DB_1 - t_s}{DB_2 - t_s} \qquad (3)$$

The coefficient of mass transfer is defined in an analogous manner with a driving potential characteristic of the mass-transfer process. There being no agreement on the best parameter for the mass potential, the most convenient, the humidity ratio difference was used. The mass transfer coefficient is defined by:

$$\frac{h_D A}{M_a} = \ln \frac{W_1 - W_s}{W_2 - W_s} \qquad (4)$$

The coefficients of heat transfer for these data were statistically correlated as a function of the coil face velocity giving the equation:

$$h = 0.616\, V^{0.5} \qquad (5)$$

The experimental heat transfer coefficient for the data is specified by Equation 5 with a standard error of estimate of 0.650 or it is within $\pm 6\%$.

The coefficient of mass transfer determined in a like manner as a function of the face velocity is:

$$h_D = 1.313\, V^{0.4} \qquad (6)$$

The mass transfer determination was not as accurate as the heat transfer due to the more difficult measurements. Equation 6 gives the experimental mass transfer coefficient within $\pm 10\%$.

It is of interest to consider the value of the Lewis relation resulting from the experimental data. The Lewis relation is

$$\frac{h}{h_D c_s} = \text{constant} \tag{7}$$

The data for this bare coil gives a ratio of heat to mass as a function of the coil face velocity. Using the mean value for the humid specific heat for all data at 0.25 and Equations 5 and 6 gives Lewis relation:

$$\frac{h}{h_D c_s} = 1.877\, V^{-.1} \tag{8}$$

The use of a mean humid specific heat for any given point of data introduces an error of less than 3%. Equation 8 represents the bulk of the data within 10%. Some points of data, however, fall outside this range since errors occurring in the coefficients in opposite directions cause additive error in calculating the value of the Lewis relation.

The state of the air at any point, x, in the coil can be determined by the following equation when the surface temperature is constant:

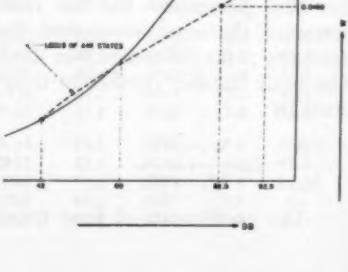
$$DB_x = t_s + (DB_1 - t_s) \times \left(\frac{W_x - W_s}{W_1 - W_s}\right)^{\left(\frac{h}{h_D c_s}\right)} \tag{9}$$

The derivation of this equation, similar to Bull,* is given in the Appendix. The exponent of the term for the water vapor content is the Lewis Relation (L.R.). If the L.R. equals one, the locus of the air states through the coil is a straight line. When L.R. is greater than one, the locus of states is a convex curve crossing the saturation line before reaching the surface temperature. For L.R. less than one, the resulting process curve is concave and always below a straight line connecting the initial air state and the coil surface temperature on the saturation line.

The experimental data presented here for the bare coil give a L.R. always greater than one. The L.R. would be one at a face velocity of 540 fpm for this coil. The data range covered was from approximately 125 to 450 fpm which gave a L.R. range of 1.16 to 1.02.

It is important to notice that a L.R. of one for a bare coil is not probable at air velocities below 500 fpm. An air velocity of approximately 1000 fpm is required for L.R. of one for small diameter single cylinders allowed to transfer

Fig. 3 Psychrometric chart plot for run No. 2 entering 92.9 F D.B. leaving 82.5 F D.B. and 42 F surface temperature



heat by both radiation and convection. The proper operating point for sling psychrometers is, therefore, around 1000 fpm.[10] The radiation shielding in aspirating psychrometers[11] permits lower operating air velocities.

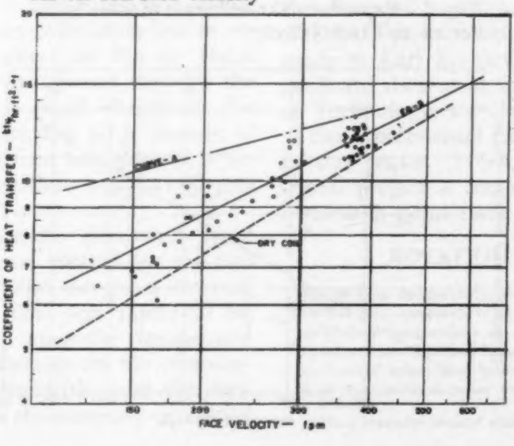
SURFACE TEMPERATURE

The difference between the actual coil surface temperature and the hypothetical surface temperature used in the contact-mixture principle (by-pass factor method) or the humidity-ratio method should be evident. If the Lewis Relation is one, the actual temperature at the surface of the water film on the coil is the same as the surface temperature given by the contact-mixture principle proposed by W. H. Carrier. This would occur at one operating velocity. At other velocities two errors enter coil calculations based on either of these two methods. The calculation of coil tests from the air side data alone results in a hypothetical surface temperature. The calculation of the heat transfer coefficient from this hypothetical temperature gives a fictitious value.

A graphical picture of this error in temperature is shown in Fig. 3, where the data from a representative run, No. 2, are plotted on a psychrometric chart. The entering, leaving and surface temperature states are plotted. The locus of the leaving conditions for a similar coil of increasing depth is plotted also. When the coil becomes infinite in depth the leaving state will be the coil surface temperature. The surface temperature given by the contact-mixture method is approximately 60 F and that by the humidity ratio method 62 F.

Given sufficient surface in depth it is noted that the air is

Fig. 4 Experimental coefficients of heat transfer vs. coil face velocity



cooled beyond the saturation line or into a "fogged" state. This condition is likely at low velocities and high Lewis numbers as approached in natural convection cooling. The phenomenon of fog dropping from a refrigerant suction line on pump-down represents this condition.

COEFFICIENTS

A plot of the experimental coefficients of heat transfer is given in Fig. 4. The solid line is the curve given from Equation 5.

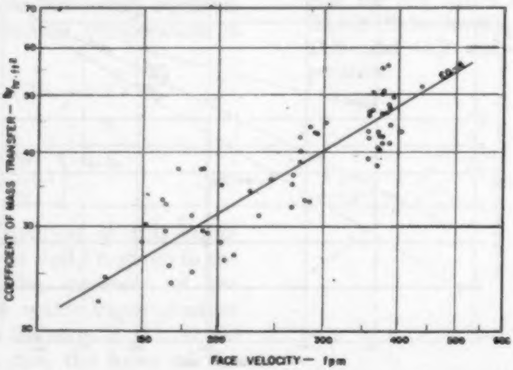
Tests were made for this coil under conditions of sensible heating and cooling without dehumidification. The dotted line on Fig. 4 represents these data. The values for the dry-coil coefficients correspond to similar data by others, for example, Kays.[12] Since the air turbulence is greater during dehumidification, it was expected that the heat transfer coefficient for dry-cooling would be lower.

If the heat transfer coefficient had been calculated without knowledge of the coil surface temperature curve "A," Fig. 4 would have resulted. This is an approximate calculation of the coefficient by the contact mixture principle. This curve crosses the experimental curve at a Lewis Relation of one, approximately 540-fpm air velocity.

The mass transfer coefficient is given in Fig. 5 and the solid line represents the statistical average curve for the data, Equation 6.

In considering these data for analysis it was found more accurate and convenient to work with the summary equations. These equations represent the best statistical correlation of the data for heat and mass transfer. The correlation of the total data showed a high degree of accuracy, although the difficulties of accurately measuring the heat and especially the mass transfer was evident in sev-

Fig. 5 Experimental coefficients of mass transfer vs. coil face velocity



eral individual points of data.

A further presentation of the data in generalized coordinates was not included here since the temperature dependence of the coefficients is small and the evaluation of the coefficient of molecular diffusion and thermal conductivity for the air-vapor mixture is somewhat a matter of conjecture.

CONCLUSIONS

It was the purpose of this paper to present experimental data for the heat and the mass transfer on a bare tube surface coil. The addition of the experimentally determined coil surface temperature provided a direct evaluation of the heat and mass transfer coefficients. This measured surface temperature was found to be much lower than would be expected from the present methods for surface temperature calculations. These differences have been explained by the variation with air velocity of the dimensionless ratio designated the Lewis Relation.

The Lewis Relation has an exponential effect on the air states as the air progresses through the coil. For a small change in the Lewis Relation, a large change of the coil surface temperature is required to produce a given coil performance.

The results presented were tabulated and plotted for the coil face velocity at the given entering air state. This was found to be satisfactory, since the dependence of the coefficients on the temperature was negligible over the data range. Thus the summary equations for the coefficients (Equations 5 and 6) were given as a function of the face velocity. For analysis, these equations give the results in a more useful form than the points of data in Table I. These correlating Equations, 5 and 6, can be used in deriving the Lewis Relation, Equation 8.

The coefficient of heat transfer for dry cooling was found to be lower than when the coil was operated with dehumidification. One would expect that the friction factor was lower also for dry cooling. Unfortunately, the core pressure drop taken with these data was quite small and the accuracy of the measurement was not sufficient for conclusions.

This research project is continuing and similar data for an extended surface coil with the addition of friction data should be available in the near future.

ACKNOWLEDGMENT

The author gratefully acknowledges the financial assistance from the ASHRAE. Acknowledgment also is made to Karl Ko-Yi Chen, former graduate student at Case Institute of Technology, for obtaining most of the experimental data presented in this paper. Work on this research project is continuing under a research grant from ASHRAE.

NOTATION

DB Dry bulb temperature of the air, F
WB Wet bulb temperature of the air, F
t_s Surface temperature of the coil, F
Q_s Rate of sensible heat transferred, Btu/hr
Q_L Rate of latent heat transferred, Btu/hr
M_v Rate of mass of water transferred, lbm/hr
A Heat transfer area of coil, ft²

h Coefficient of heat transfer from air to surface, Btu/hr ft² F
M_a Mass rate of flow of "dry" air, lbm/hr
C_h Humid specific heat, Btu/lbm R
l_n Napierian or natural logarithm
h_D Coefficient of mass transfer, lbm/hr ft²
W Humidity ratio, lbm water/lbm air
V Coil face velocity of the air, fpm
L.R. Lewis Relation, $\frac{h}{h_D c_h}$
λ Latent heat of vaporization, Btu/lbm

REFERENCES

1. W. H. Carrier, Contact Method of Determining Finned Coil Capacities, Heating, Piping and Air Conditioning, Dec. 1936.
2. G. L. Tuve and L. G. Seigel, Performance of Surface-Coil Dehumidifiers for Comfort. Trans. ASHVE, Vol. 44, 1938.
3. W. K. Lewis, Evaporation of a Liquid into a Gas, Trans. ASME, Vol. 44, 1922.
4. W. H. Carrier, Rational Psychrometric Formulae, Trans. ASME, Vol. 23, 1911.
5. C. D. Ware and T. H. Hacha, Heat Transfer from Humid Air to Fin and Tube Extended Surface Cooling Coils, ASME paper 60HT17, June 1960.
6. E. R. G. Eckert, Heat and Mass Transfer, McGraw-Hill Book Co. (1959).
7. W. McAdams, Heat Transmission, McGraw-Hill Book Co., 3rd Ed. (1955).
8. W. Goodman, Performance of Coils for Dehumidifying Air Heating, Piping and Air Conditioning, Nov. 1938 to May 1939.
9. L. C. Bull, Coils for Cooling and Dehumidifying Air, Journal I.H.V.E., April 1959.
10. W. H Walker, W K. Lewis, W. H. McAdams and E. R. Gilliland, Principles of Chemical Engineering, McGraw-Hill Book Co., 3rd Ed. (1937).
11. F. M. Flanigan, Humidity Measuring Instruments, ASHRAE JOURNAL, Dec. 1960.
12. W. M. Kays and A. L. London, Compact Heat Exchangers, McGraw-Hill Book Co. (1958).

APPENDIX

Over a differential element of a coil surface, the air changes temperature by ΔDB and at the same time is reduced in moisture content by ΔW. The total heat transferred to a surface element, dA, is the sensible heat from the air and from the water vapor in the air and the latent heat equivalent of the water condensed from the air

$$Q_T = Q_s + Q_L \qquad \text{(a)}$$

The sensible heat transferred is

$$Q_s = hdA\,(DB - t_s) = -M_a c_h\, dDB \qquad \text{(b)}$$

The humid specific heat, c_h, is used to account for both the cooling of the air as well as for the cooling of the water vapor in the air. The desuperheating of the water condensed on the differential element is

$$M_a c_v (DB - t_s)\, dW$$

and the equation for the sensible heat transferred at the differential element is:

$$h\,(DB - t_s)\,dA = -M_a c_h\, dDB - M_a c_v (DB - t_s)\, dW \qquad \text{(c)}$$

The latent heat or mass of water condensed on the differential element is

$$\lambda Q = h_D (W - W_s)\, dA = -M_a\, dW \qquad \text{(d)}$$

By combining equations c and d the ratio of the mass change to temperature change can be obtained

$$\frac{dW}{dDB} = \frac{c_h}{(DB - t_s)\,\frac{h}{h_D} \cdot \frac{1}{W - W_s} - c_v} \qquad \text{(e)}$$

For a constant surface temperature this equation can be integrated over the total coil surface. Note that the specific heat of the superheated steam in the air c_v can be represented as

$$c_v = \frac{c_h - c_a}{W} \qquad \text{(f)}$$

Making this substitution and integrating from initial state 1 to final state 2 gives:

$$\frac{h}{h_D c_h} l_n \left(\frac{W_2 - W_s}{W_1 - W_s} \right) - \left(\frac{c_h - c_a}{c_h} \cdot l_n \frac{W_2}{W_1} \right) = l_n \left(\frac{DB_2 - t_s}{DB_1 - t_s} \right) \qquad \text{(g)}$$

It can be seen that the second term on the left contributed by the desuperheating of the condensed water is

negligible. Making this simplification, the equation reduces to:

$$\frac{DB_2 - t_s}{DB_1 - t_s} = \left(\frac{W_2 - W_s}{W_1 - W_s} \right)^{\frac{h}{h_D c_a}} \qquad (h)$$

It is to be noted here that the ratio of the temperature differences is equal to the ratio of the humidity ratio differences to an exponential power. The exponent is the Lewis Relation. The locus of the air temperature at any point, x, in the coil is then given by Equation 9 in the paper.

DISCUSSION

J. B. CHADDOCK, Lafayette, Ind.: It is stated that the Lewis Relation as applied to heat and mass transfer problems, in solving problems involving air and water vapor mass problems, is a debatable point. It is indicated that the Lewis ratio is one that has not been firmly established. I disagree with this.

Secondly, on the basis of the experiments, it is stated that the heat transfer coefficient is proportional to the velocity of five-tenths power and the mass transfer to the velocity of six-tenths power, indicating that velocity affects these two things differently. It seems to me that this is in error. It is well known that there is an analogy between heat and mass transfer. With regard to conduction heat transfer, and this would exist close to the surface where the fluid would be in a laminar flow, then it would be the thermoconductivity times the mass transfer. Similarly, the same form is seen for the laminar boundary equations. It is more difficult to see the analogy in turbulent flow, but it does exist there. The point is that many experiments carried out over a number of years have shown quite conclusively, I think, that this analogy is valid, provided that the mass transfer rate is low. In all air conditioning problems this mass transfer rate is quite low and, as a result, it can be said that this analogy is true.

Now, the analogy takes many different forms. One of the most common of these relates the J factors. If this is done, then it is seen that the Lewis Relation for air and water vapor turns out to be equal to the Schmidt, and there is no question that this analogy is quite valid. Therefore, I question this as a debatable point, that velocity can affect the heat and mass transfer coefficients in different ways.

Certainly the problem of collecting heat and mass transfer data is an extremely difficult one. One of the problems is to maintain a surface completely so that the heat transfer and mass transfer areas are identical and it can be concluded that the coefficients are based on the same area.

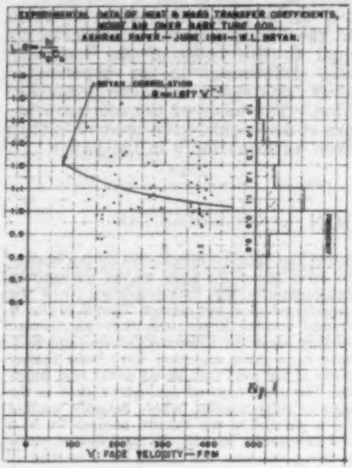
AUTHOR BRYAN: When we started with heat transfer, the first analogy we had to make was the analogy between heat transfer and fluid friction; the classical work of Reynolds and later by Blasius and Pohlhausen on the flat plate corresponds to the foundations that we use in these analogies. These works also point out where it is possible to go astray in that if the velocity distribution in the boundary-layer is not the same as the temperature distribution in the boundary-layer, then the simple analogy is not correct. That is to say, if Prandtl number is not one, then our friction analogy has to be corrected. These corrections have been made and they have been further made as we went into liquid metal heat transfer. The same type of analogies follow in the case of mass transfer, where Schmidt number has a parallel to the Prandtl number for heat and momentum transfer. So these factors, while pointing the way and giving us a method of calculating or making a good substantial guess at the coefficients of heat transfer and the coefficients of mass transfer, certainly cannot be assumed to be accurate in all cases.

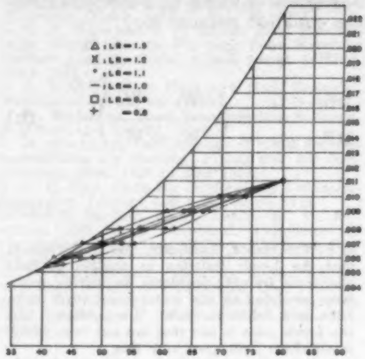
T. KUSUDA, East Orange, N. J. (Written): Professor Bryan has correlated his experimental data of heat and mass transfer coefficients on dehumidifying bare tube coils with respect to air velocity. The statistical curve fitting of these two coefficients has resulted in Equations (5) and (6). By combining these two equations, he has attained Lewis Relation as a function of air velocity as shown in Equation (8).

The values of Lewis Relation have been computed at each test run result of Professor Bryan's and plotted against air velocity. Fig. 1 shows the results of these individual plots superimposed by the Bryan relation (Equation 8).

The nature of the individual plots which do not seem representing Professor Bryan's statistical equation are somewhat disturbing. If I were to choose the value of Lewis Relation from the plot, I would take the value that represents the highest value of the frequency curve, which is 1.1 as shown at the right hand side of the chart. But I would still be reluctant to deny the validity of LR = 1 in view of the data scattering.

Furthermore, Fig. 2 illustrates the dry bulb and specific humidity curve (or coil curve) on a Carrier psychrometric chart for various Lewis numbers when entering air is of 80 DB and 67 WB and surface temperature is 40 F. These curves are plotted using Equation (9) of the paper. As indicated, the air state is no longer represented by a straight line connecting the entering air state and surface temperature if the Lewis number differs from unity. But the deviation is small in the case of Lewis number 1.1.





Three things to be considered with regard to this air state curve are: If the surface temperatures were to be determined from the entering and leaving air space states of a shallow curve, this slight curvature of the state line would lead to an appreciable error.

If the surface temperature is known, the leaving airstate can be determined by the straight line relation without losing engineering accuracy even if the state line is nonlinear.

The non-linearity of the air state line on the psychrometric chart has been pointed out previously. But it should be realized that so far the nonlinearity is attributed mostly to the variation of surface temperature itself from row to row; not to the Lewis Relation. This point could have been contested if the author had tested much deeper coils with constant coolant temperature instead of the constant surface temperature, and measured the surface temperature row by row. The surface temperature determination becomes a major problem not when the in and outlet air states are known, such as is the case of the author's experiment, but when we are confronted with determining the coil capacity with the knowledge of inlet conditions of air and coolant alone. This problem is crucial to our industry in rating the dehumidifying apparatus.

L. F. Albright, Lafayette, Ind.: In Figs. 4 and 5, where the coefficients of heat transfer and mass transfer were plotted vs. the face velocity only, the obvious face velocity is important and the minimal number and Schmidt number, etc. go into this. What is complicating, is the fact that both heat and mass transfer occur simultaneously and that this is an important variable, too; the amount of mass transfer, for example, would affect the heat transfer. Would it be possible for you to go back in your data and to recorrelate and see, with a given amount of mass transfer, whether these could be straightened out and to find out how much effect the amount of mass transfer, for example, would have on the heat transfer coefficient?

Author Bryan: The heat transfer and mass transfer coefficients were calculated separately. The quantity of heat determined by mass transfer is given in the tabulated data in both instances—the heat and the mass transfer.

If we assume the statistical plot, the first curve made by Mr. Kusuda, and that Lewis Relation 1 as he calculated it, then the coefficient of heat transfer could be calculated from his statistical plot and a different curve would be obtained for the heat transfer coefficient predicted from Lewis Relation equal 1. This would give a curve that would fall well below all of the data that were taken for the heat transfer coefficient.

The difficulty here may not be quite apparent, but it is this. In these experiments, the surface temperatures were measured and the quantities of heat were measured. Therefore, the coefficient of heat transfer and the coefficient of mass transfer were calculated from the fundamental data. Also, from the fundamental data, we can calculate the ratio of the Lewis Relation for each specific piece of data, and this specific piece of data would then be a secondary calculation. It would carry two errors involved with it: the first experimental error and the second with the error advanced when we calculated the coefficient and when we calculated the relation from the coefficient ratios. If the heat transfer has an error in one direction and the mass transfer in another direction, then these are

compounded in the Lewis Relation. Therefore, it is better to take the statistical curve of the heat transfer and the statistical curve of the mass transfer and then the Lewis Relation just turns out as a ratio of these two. The ratio of the individual data points sometimes amplifies the error when it is in opposite directions.

DONALD D. RICH, Syracuse, N. Y. (Written): Combined heat and mass transfer in surface coils is one of the most fundamental processes in air conditioning. The problems involved in experimentally measuring the heat transfer rate, mass transfer rate, and particularly the surface temperature, are not simple, and the author is to be commended for his efforts on this difficult problem.

In reading the paper, I was somewhat surprised to find the statement, "the assumption that this relation (the Lewis Relation) applies to dehumidifying coils has not been proven correct." Although it is known that the Lewis Relation is not exact, there have been considerable data taken on commercial coils which indicate that, in practice, fairly accurate results are obtained using this Relation. Recently, in fact, the ARI committee for revision of the standard for Forced Circulation Air Cooling and Air Heating Coils, has analyzed over 200 test points taken on more than 20 wet surface coils, and has concluded that the enthalpy potential method, which is based on the Lewis Relation, predicts coil performance quite satisfactorily.

It would appear to me that any attempt to calculate the surface temperature, or from it the heat transfer coefficient, using only the entering and leaving air conditions, is bound to be highly unreliable, even if the correct Lewis Number is known. The reason is that even a small experimental error in either the entering or leaving air conditions will be greatly magnified in extrapolation to the saturation line in order to determine the surface temperature. This effect becomes more and more pronounced as the change in air conditions decreases.

It would be interesting to know whether measurements were made of the water flow rate and temperature. These data would be valuable in two aspects. First, it would be possible to calculate the water side heat transfer coefficient, and, from this, the tube surface temperature, which could then be checked against the measured value. Secondly, a computation of the total capacity, based on the water side measurements, could be used as a check on the air side measurements. Along the same lines, it would be interesting to know how closely the condensation rate measurements checked the air humidity difference.

1755

No. 1755

A Study of Fluid Flow through Flexible Orifices

R. D. ULRICH W. E. FONTAINE O. W. WITZELL
Member ASHRAE

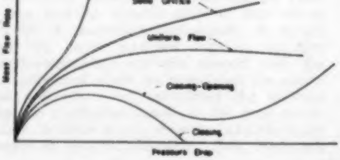
A "flexible" orifice is one in which the orifice area varies with pressure. Flexible orifices are of interest because of the large variety of flow rate-pressure drop relations which may be obtained by various designs of the orifice and the method of supporting the orifice (see Fig. 1). The phenomenon of decreasing flow rate with increasing pressure drop suggests the use of these orifices as devices for the regulation of flow. Attempts have been made to use these orifices in refrigeration systems to yield maximum flow rates at design conditions and to decrease the flow rate with increasing pressure differences.

Holman[1] investigated this problem by comparing power consumption in a refrigeration system with expansion occurring through capillary tubes and through flexible orifices. The orifices used by Holman were designed to provide a constant flow rate irrespective of the pressure difference. Holman observed that these uniform flow orifices required less power input than that required with capillary tubes at high condenser pressures.

The objective of the present work was to determine the variables which influenced the flow. After these variables had been determined, the problem was then one of correlation.

The geometries considered in

Fig. 1 Qualitative flow characteristics observed



R. D. Ulrich is Associate Professor of Mechanical Engineering at Brigham Young University. W. E. Fontaine and O. W. Witzell are Professors of Mechanical Engineering at Purdue University. This paper was prepared for presentation at the ASHRAE 68th Annual Meeting, June 26-28, 1961, Denver, Colo.

this work were short, right cylindrical orifices with centered, cylindrical holes. The orifices were made of rubber. The support plates were flat with circular holes at least as large as the orifice. The fluid flowing was water at room temperature. Only steady flow was studied.

THEORETICAL CONSIDERATIONS

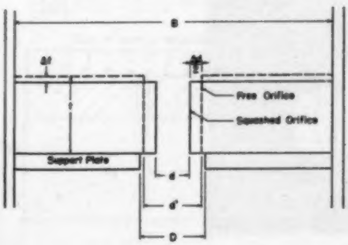
The steady flow continuity equation as applied to a flexible orifice yields the mass flow rate as

$$M_m = \rho VA \qquad (1)$$

where M_m is the mass flow rate, ρ the fluid density, A the minimum area of the hole and V the average velocity at that section. Hence, the problem becomes one of determining A, or d the minimum diameter, and V.

Development of formula for orifice diameter—The diameter which was considered was the minimum orifice diameter under a loaded condition. Three effects were considered to influence the change in minimum diameter. They were the "squash" effect, or the effect of axial compression, the "twist" effect, or the effect of rotating the nonsupported portion of the flow washer about the rim of the support plate, and the effect of large changes in diameter of elements of the orifice. The latter effect is due to the build up of hoop stresses causing deviation from the "small deflection" theory.

Fig. 2 Cross-section of orifice showing squash effect



The squash effect was developed based upon the following assumptions: (1) constant volume of rubber, (2) no change in outside diameter, (3) the change in thickness Δt is small compared to the thickness t and (4) only the supported portion of the washer deforms axially (see Fig. 2). The above assumptions lead to

$$\frac{\pi}{4}(B^2 - D^2)\,\Delta t = \frac{\pi}{4}(d'^2 t - d^2 t) \qquad (2)$$

Equation 3 will be used to define a compression modulus similar to Young's Modulus, which will be called the effective modulus for squash, E_s,

$$\frac{\Delta t}{t} = \frac{\Delta P}{E_s} \qquad (3)$$

where ΔP is the uniformly distributed load on the orifice. Combining Equations 2 and 3 yields

$$(B^2 - D^2)\frac{\Delta P}{E_s} = d'^2 - d^2 \qquad (4)$$

or Equation 4 may be written in the form

$$\Delta d = d'\left[1 \pm \sqrt{1 - \frac{\Delta P}{E_s}\,\frac{B^2 - D^2}{(d')^2}}\,\right] \qquad (5)$$

where Δd is the change in minimum diameter. The positive radical

has no meaning in this problem, since it would indicate a Δd when ΔP is zero, an impossible situation. It was desirable to non-dimensionalize Equation 5 by dividing each side by 2t in order to get the squash effect on the diameter in the same form as the twist effect. Thus, Equation 6 is the final form for the squash effect.

$$\frac{\Delta d}{2t} = \frac{d'}{2t}\left[1 - \sqrt{1 - \frac{\Delta P}{E_s}\,\frac{B^2 - D^2}{(d')^2}}\right] \qquad (6)$$

The twist effect was developed by considering the portion of the flow washer which was not supported by the back-up plate. This portion of the washer is a ring, simply supported along its outer periphery on its lower edge. It was assumed that the supported portion of the flow washer did not influence the twist effect (see Fig. 3). The determination of the angle of twist θ and the minimum hole size based on this angle and other geometric factors were studied.

For small angular deflections in a ring of rectangular cross section rotating about the geometric center of the rectangle, Timoshenko[3] gives the formula

$$\theta = \frac{12M}{Et^3 \ln \dfrac{D}{d'}} \qquad (7)$$

where M is the twisting moment uniformly distributed around the ring and E is Young's Modulus. Equation 7 results from the integration of a differential relation between θ and t. When the rotation is about the peripheral corner instead of the center of the rectangle, the limits of integration are changed, resulting in

Fig. 3 Cross-section showing twist effect

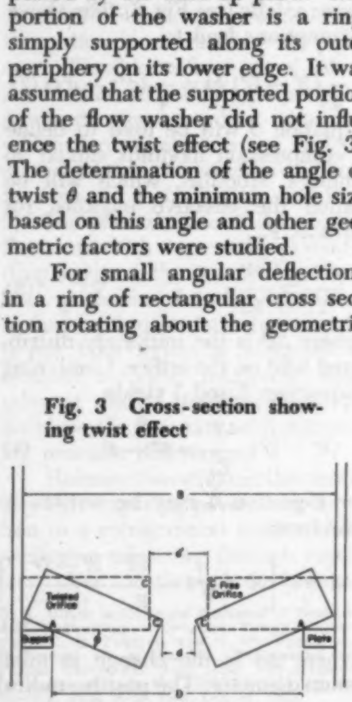
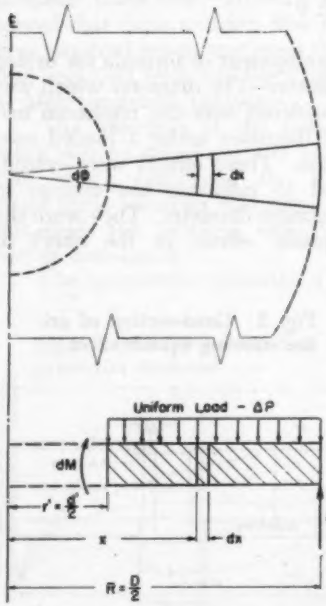


Fig. 4 Sketch for deriving the uniformly distributed moment M



$$\theta = \frac{3M}{E_t t^2 \ln \frac{D}{d'}} \tag{8}$$

Equation 8 here is used to define E_t, the effective modulus for twist. It is assumed that this equation is valid for large angular deflections.

The moment M was calculated by assuming the ring to have a uniform load ΔP distributed across its top surface, and no other loads were present except the simple support along the lower periphery (see Fig. 4). The moment dM on the element $d\phi$ is equal to the moment of the load dM_L minus the moment of the support dM_F, or

$$dM = -dM_L + dM_F \tag{9}$$

Consider first dM_L,

$$dM_L = \Delta P \, d\phi \int_{r'}^{R} x\,(x - r')\,dx \tag{10}$$

or $dM_L =$

$$\Delta P \, d\phi \left[\frac{R^3}{3} - \frac{r' R^2}{2} - \frac{r'^3}{3} + \frac{r'^3}{2} \right] \tag{10a}$$

Now consider dM_F,

$$dM_F = dF\,(R - r') \tag{11}$$

or

$$dM_F = \Delta P \frac{(R^2 - r'^2)}{2} d\phi\,(R - r') \tag{11a}$$

Integrating $d\phi$ for ϕ varying from 0 to 2π and substituting into Equation 9 yields

$$\frac{M}{2\pi\,\Delta P} = \frac{R^2 - r'^2}{2}(R - r') - \left[\frac{R^3}{3} - \frac{r'^3}{3} + \frac{r'^3}{3} - \frac{r' R^2}{2} \right] \tag{12}$$

Equation 12 may be reduced to

$$M = \frac{\pi}{8} \Delta P\, D^3 \left[\frac{1}{3} - \frac{d'^2}{D^2} + \frac{2}{3}\frac{d'^3}{D^3} \right] \tag{13}$$

Substituting Equation 13 back into Equation 8 gives

$$\theta = \frac{\Delta P}{E_t} \frac{D^3}{t^2} \frac{\pi}{4 \ln \frac{D}{d'}}$$

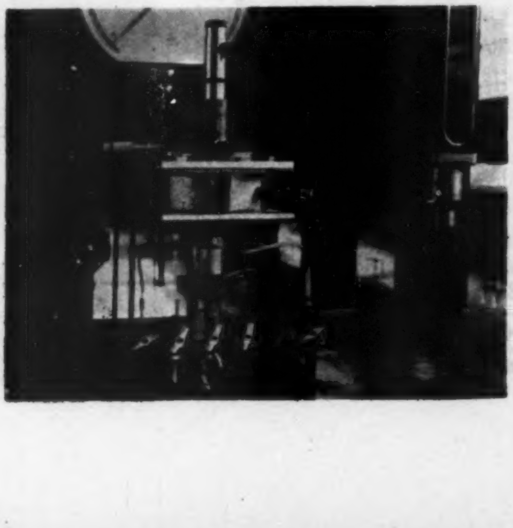


Fig. 5 Photograph of test section

$$\left[\frac{1}{2} - \frac{3}{2}\frac{d'^2}{D^2} + \frac{d'^3}{D^3}\right] \qquad (14)$$

which is the final equation for the angular twist in radians.

The twist effect was considered to depend upon the angle of twist θ and geometric factors. The assumptions were that during deflection the rectangular cross section did not deform and that each rectangular section rotated about the point of contact with the support. These assumptions lead to point C rotating on the arc of a circle having its center at A and radius $\overline{AC}$. The diameter of the hole at C is the minimum diameter at any angular deflection θ.

The change in radius as a function of θ is given by the equation

$$\Delta r = AC \cos(\beta - \theta) - (R - r') \qquad (15)$$

where β is the angle between the diagonal and the base of the rectangle. Thus

$$\beta = \text{arc tan} \frac{t}{R - r'} \qquad (16)$$

Expanding cos $(\beta\text{-}\theta)$ and using the identities $\cos \beta = \dfrac{R\text{-}r'}{\overline{AC}}$ and sin $\beta = \dfrac{t}{\overline{AC}}$ yields the expression for Δr,

$$\Delta r = (R - r')(\cos \theta - 1) + t \sin \theta \qquad (17)$$

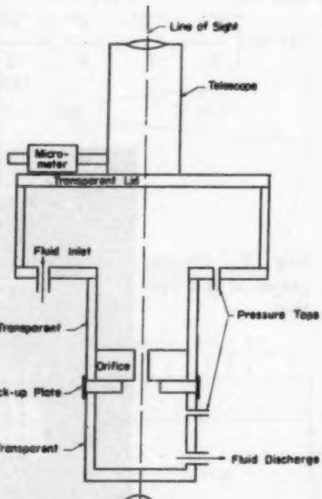
Multiplying Equation 17 through by 2 and then dividing each term by 2t yields the final expression

$$\frac{\Delta d}{2t} = \frac{(D - d')}{2t}(\cos \theta - 1) + \sin \theta \qquad (18)$$

Equation 18 gives the dimensionless change in diameter as a function of θ and geometric properties.

Effect of large strains on the diameter change — The assumption of small deflections, both linear and angular, has been made. The maximum unit strain was on the order of 50%. The maximum angular deflections were 30 to 50 deg. Thus, it is apparent that this assumption is not valid. The main effect of the large deflections is to cause distortion of the rectangular cross section of the flow washers. In each case, squash and twist, the effect is to cause the change in diameter of the flow washers to

Fig. 6 Schematic diagram of test section

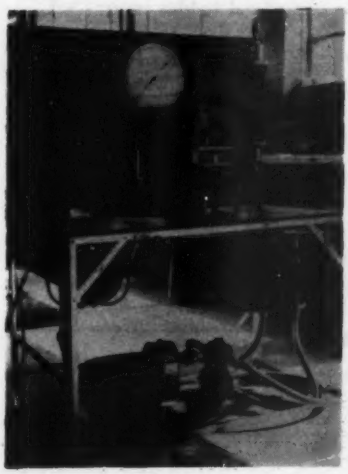


counteract the combined effort of squash and twist so that the change in diameter is less than expected.

In the case of squash, as the hole becomes smaller the rubber immediately surrounding the hole gets stiffer until, in the limit where the hole vanishes, the washer is essentially rigid. In the case of twist, the lateral forces between the elements induce stresses which cause the rectangle to distort. The compressive stresses above the neutral surface appear to stiffen the rubber near the hole and thus restrict deformation.

This entire effect is a function of the stress distribution in the orifice. Due to the complexities involved in dealing with large deflections and with rubber, this effect was determined experimentally and will be discussed in that section of

Fig. 7 Photograph of apparatus



the paper. The effect appears there as a multiplying correction factor to the squash and twist effects.

To obtain the minimum diameter under loaded conditions, or the change in diameter due to loading, the squash effect and the twist effect are added and the sum is multiplied by the "large deflection" correction factor S. Hence, the final equation for the dimensionless diameter change is

$$\frac{\Delta d}{2t} = S\left\{\frac{d'}{2t}\left[1-\sqrt{1-\frac{\Delta P}{E_s}\,\frac{B^2-D^2}{(d')^2}}\right] + \frac{(D-d')}{2t}(\cos\theta-1)+\sin\theta\right\} \tag{19}$$

To determine velocity – The conservation of energy and mass equations for steady incompressible flow when applied to a frictionless process for cases similar to this one yield

$$V'_2 = \frac{\sqrt{2g\dfrac{\Delta P}{\omega}}}{\sqrt{1-\left(\dfrac{A_2}{A_1}\right)^2}} \tag{20}$$

where the subscripts 1 and 2 refer to the upstream and minimum diameter locations respectively, and the prime (′) indicates ideal conditions. Since friction was involved, the velocity was multiplied by a discharge coefficient C. The flow coefficient K was used where

$$K = \frac{C}{\sqrt{1-\left(\dfrac{A_2}{A_1}\right)^2}} \tag{21}$$

and was in all cases almost equal to C since A_2 was much less than A_1. Thus, the expression for velocity at the smallest cross section of the hole was

$$V = K \sqrt{2g \frac{\Delta P}{\omega}} \qquad (22)$$

The flow coefficients were determined experimentally and will be discussed presently.

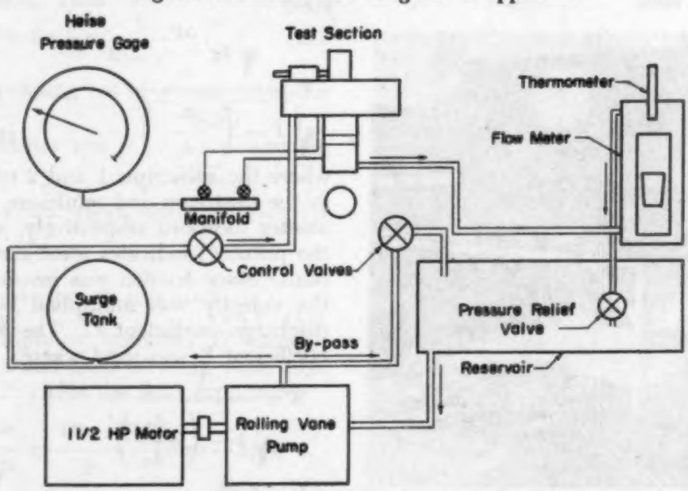
APPARATUS AND METHODS

In order to observe the orifice visually, the supporting section for the flow washer was made of plexiglass. See Fig. 5 for a photograph of the test section and Fig. 6 for a schematic sketch of the test section. For checking the theoretical value for the minimum diameter of the hole and to determine the flow coefficients, the following measurements were taken at each steady state condition: (1) Minimum diameter. (2) Pressure difference across the orifice. (3) Mass flow rate. (4) Fluid temperature.

Diameter—The minimum diameter was measured by traversing the hole with a traveling telescope mounted on top of the test section. The vertically mounted telescope was driven in the horizontal direction by a depth micrometer having at least a count of 0.001 in.

Pressure Difference—Both the upstream and downstream pressures were measured with the same pressure gage. The upstream pressure tap was in the entrance box about two tube diameters from the top surface of the orifice. The downstream pressure tap was located

Fig. 8 Schematic flow diagram of apparatus



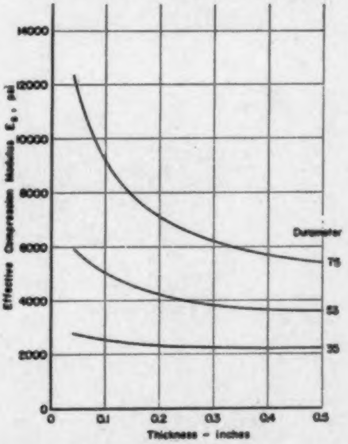
about one-half tube diameter from the support plate. These two pressure taps were connected to a selection manifold which was connected to the gage. The gage range was from 0 to 500 psi with a least count of ½ psi.

Mass Flow Rate – The flow rate was measured by a flow meter which had been calibrated in place. The range of the flow meter was from 0 to 27 lb water/min with a least count of 0.1 lb/min.

Temperature – The temperature was measured with a mercury-in-glass thermometer mounted in a well on the top of the flow meter. In the range of temperatures occurring in the tests, the thermometer least count was 1 F.

Other Apparatus – The remaining apparatus consisted of a rotating vane pump, surge tank and a reservoir designed to deliver water to the orifice under steady state conditions. The water was treated with a minute amount of sodium dichromate to resist corrosion. See Fig. 7 for a photograph of the apparatus and Fig. 8 for a schematic flow diagram.

Fig. 9 Effective compression modulus E_s versus thickness for various rubber hardnesses



Testing procedure for one datum point consisted of setting the flow rate at some particular value and, after a nominal period of time for steady state to occur, the four items of interest—pressure drop, flow rate, hole diameter and temperature—were recorded. From five to twenty such points were taken for each test setup. In each case the pressure drop for one point was always greater than for the preceding point. This was to eliminate the additional variable associated with the hysteresis in the stress-strain relationship inherent with rubber. Experimental information that could be obtained was limited by these conditions:

1. The maximum pressure of the pump (approx. 250 psig) was obtained.
2. The maximum flow meter capacity (approx. 25 lb per min) was obtained.
3. Under certain conditions cavitation in the orifice occurred.
4. Under certain conditions buckling of the rubber caused the orifice to deviate from a circular configuration. At some of the cited conditions the hole was observed to vibrate. In all cases the end

result of buckling was the collapse of the hole.

EXPERIMENTAL RESULTS AND DISCUSSION

The main problem concerning the squash effect was the evaluation of the Effective Modulus for Squash E_s. Unlike most other substances common in engineering use, the modulus for rubber in compression is a function of geometry as well as hardness. Most authors [3,4,5,6] on rubber properties consider the effective modulus as the product of the small deflection, unit cube modulus and a shape factor. The shape factor is defined as the area of one loaded surface divided by the total area of the free surfaces.

One author [7] suggests that the shape factor and thus the effective modulus is a function mainly of thickness. There appears to be an interdependence between hardness and geometry. The reason for this dependence is due to the inability of the loaded surfaces to deflect tangentially to the load. It is shown in (5) that when both of the loaded surfaces were well lubricated the compression modulus no longer depended on geometry and was constant up to about 20% deflection.

It was decided to use Equation 6 as a definition of E_s and measure Δd, d', t, ΔP, B and D and calculate E_s because: (1) The axial loadings in this work were hydrostatic on the top surface and supported by a smooth, non-lubricated, metal back-up plate, (2) no shape factors were available for the annular configuration where the outside diameter was fixed and (3) axial deflections were much less than 20% but the maximum radial deformations were about 50%. The results of the calculations are shown in Fig. 9.

These tests were run with the back-up plate hole diameter the same as the free orifice diameter, thus eliminating any possible twisting effect. The curves of Fig. 9 were determined by plotting $1 - \left(\frac{d}{d'}\right)^2$ versus ΔP for each test.

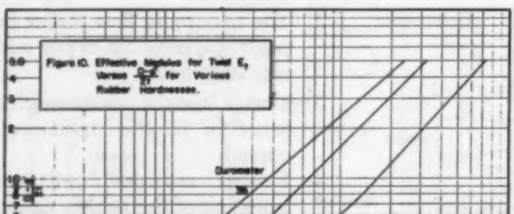


Fig. 10 Effective modulus for twist E_t versus $\frac{D-d'}{2t}$ for various rubber hardnesses

These curves were nearly straight lines up to about 20 to 25% change in diameter. The E_s values were calculated from the slopes of these straight lines. A large deflection correction S, which is discussed later, was used to determine the effective modulus at large deflections. The E_s values from Fig. 9 permit the calculation of $\frac{\Delta d}{2t}$ in Equation 6.

Twist Effect – The determination of the effect of twist on $\frac{\Delta d}{2t}$ was based upon the angular deflection θ and the experimental values of the Effective Modulus for Twist E_t. In order to evaluate E_t, Equation 14 was solved explicitly for E_t

$$E_t = \frac{\Delta P}{\theta} \frac{D^2}{t^2} \frac{\pi}{4 \ln \frac{D}{d'}} \left[\frac{1}{2} - \frac{3}{2} \frac{d'^2}{D^2} + \frac{d'^4}{D^4} \right] \tag{23}$$

Table I. Large Deflection Correction Factor, S

$\frac{\Delta d}{d'}$ Calculated[1]	0	0.2	0.4	0.6	0.8	1.0
S for 35 (Durometer)	1	0.80	0.70	0.60	0.50	0.40
S for 53 (Durometer)	1	0.85	0.75	0.70	0.60	0.50
S for 75 (Durometer)	1	0.90	0.80	0.70	0.70	0.65

[1] Calculated from the sum of Equations 6 and 18.

Geometric values and ΔP were recorded for the various flow conditions. The problem remaining was the evaluation of θ corresponding to ΔP in Equation 23. The experimental values of $\frac{\Delta d}{2t}$ were plotted as a function of ΔP. The squash effect as calculated by Equation 6 and corrected by the S factor, was subtracted from the total, leaving only the twist effect on $\frac{\Delta d}{2t}$. In order to obtain experimental values of θ to be used in Equation 23, $\frac{\Delta d}{2t}$ versus θ was plotted for various values of $\frac{D-d'}{2t}$ according to Equation 18. For the various geometric arrangements Δd was then measured for several different pressure drops. These values of Δd were used to determine θ which was used in Equation 23 along with the corresponding ΔP and geometric factors to determine E_t. These results are presented in Fig. 10. The E_t values from Fig. 10 may be used in calculating θ in Equation 14 and the θ can be used in Equation 18 for obtaining $\frac{\Delta d}{2t}$.

Large Deflection Correction – A comparison of $\frac{\Delta d}{2t}$ calculated versus

$\frac{\Delta d}{2t}$ measured indicated that the hole size did not deviate from the free hole diameter as much as calculated. This was true both for decreasing diameters and increasing diameters. One possible reason for this is that, during large deflections, the assumptions concerning small deflections and constant cross section shape are violated, and in some cases the violation was large. A correction factor was determined based upon the maximum unit deflection $\frac{\Delta d}{d'}$ and rubber hardness. These results are given in Table I, indicating that there is more deviation from the assumptions for softer rubber than for harder rubber.

In addition to this correction it was noted that in each case where the hole diameter reached a minimum with respect to ΔP and then started to enlarge as the pressure drop was increased, a straight line relationship resulted between diameter and ΔP after the minimum diameter was reached. Fig. 11 shows a qualitative comparison of these corrections. The location of the intercept of the $\frac{\Delta d}{2t}$ measured curve was very near the intercept of the calculated curve and a straight line was drawn from the point of maximum $\frac{\Delta d}{2t}$ (after corrected by multiplying by S) through the calculated intercept. This straight line may be extended to angular deflections of approximately $\theta = 2.5\ \beta$. No experiments were run at higher angular deflection.

The average difference between the diameter as calculated above and the measured diameter was about 3%. The maximum difference was 6% when the deflection was less than 40%, which is the maximum allowable before the hole buckles.

The flow coefficients were cal-

Fig. 11 Qualitative comparison $\frac{\Delta d}{2t}$ calculated and measured

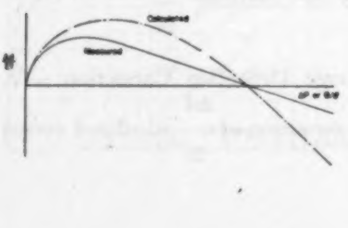
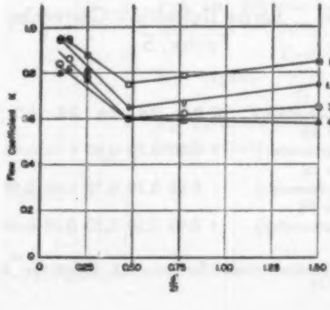


Fig. 12 Flow coefficient K versus $\frac{d'}{2t}$ for various $\frac{D}{d'}$ values and for 35 durometer orifices



culated from the equation

$$K = \frac{0.004\ M}{d^2\ \Delta P} \qquad (24)$$

where 0.004 is a constant for water at room temperature and includes unit conversions. Note that K was based upon the minimum orifice diameter.

Correlation—An attempt was made to correlate the flow coefficients with an idealized Reynolds number T[3] where

$$T = \frac{N_{RE}}{K} \qquad (25)$$

The T number in Equation 25 would allow for the direct determination of flow rates whereas the usual N_{RE} would necessitate a trial-and-error solution. It was found that K was almost independent of T for thin washers $\left(\frac{d'}{2t} \geqq 0.5\right)$. For

Table II. Effect of T Numbers on K

(Add 10% to K for each 20,000 increase of T in this range)

Durometer	T Number Range for K Constant	T Number Range for Varying K
35	15,000-25,000	25,000-50,000
53	20,000-40,000	40,000-80,000
75	18,000-40,000	40,000-80,000

thick washers $\left(\frac{d'}{2t} \leqq 0.5\right)$, the K values shown in Fig. 12, 13 and 14 were evaluated for T numbers as shown in Table II. The tendency was for the flow coefficients to be almost constant in a specific range of T numbers, as shown in Table II, and to increase linearly with T above a certain value of T.

For hard, thin washers there appeared to be a slight tendency for the K values to decrease at

[3] The use of the T number is similar to that suggested by Binder (8).

Fig. 13 Flow coefficient K versus $\frac{d'}{2t}$ for various $\frac{D}{d'}$ values and for 53 durometer orifices

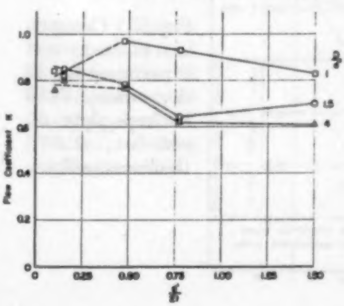
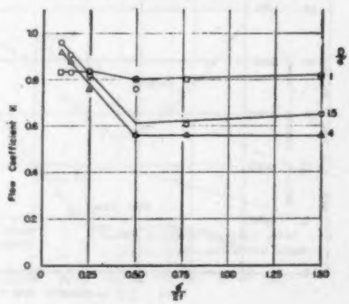


Fig. 14 Flow coefficient K versus $\frac{d'}{2t}$ for various $\frac{D}{d'}$ values and for 75 durometer orifices



higher T numbers but this was not of sufficient amount to show a valid trend.

Variables which had the most important effect on K were orifice thickness and support plate diameter. These variables were represented in dimensionless form $\frac{d'}{2t}$ and $\frac{D}{d'}$. The flow coefficients increase with thickness in the range of these tests. This may indicate that the thick orifices become more streamlined than the thinner ones. This may also be due to the downstream pressure tap being located at some point other than the vena contracta as is required by Equation 24. For thicker washers the vena contracta would be farther upstream from the downstream pressure tap and more distance would be allowed for pressure recovery so the downstream pressure would become higher. This causes ΔP to be lower, which makes K appear larger than it should be.

The wider spread of flow coefficients for soft washers having $\frac{D}{d'} < 1$ and $\frac{d'}{2t}$ from 0.5 to 1.5 than for hard washers with these same geometrics was probably due to greater deformation of the softer washers. This deformation would tend to convert the orifice into a flow nozzle.

Another observation made was that for orifices which first decreased in diameter and then increased as the pressure drop increased, K increased about 5 to 20%. The K values given on the graphs are for the decreasing diameter phenomena and a 10% increase is recommended for the flow coefficients after the orifice begins to open up. The correlation of K with geometrical properties are shown in Figs. 12, 13 and 14 for three different hardnesses.

The results of three typical experimental tests are shown in Figs. 15, 16 and 17. These tests

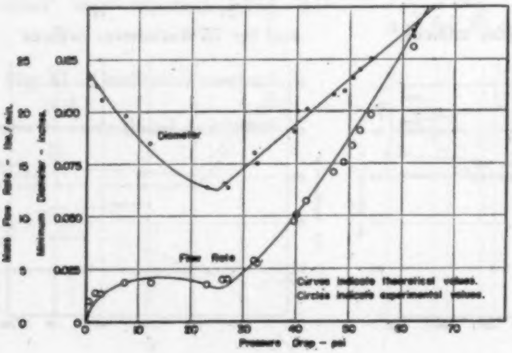


Fig. 15 Comparison of theory and experiment for 35 durometer, 7/16 backup plate diameter, 0.075 thickness orifice

were picked at random from a large number of similar tests. The results indicate very good prediction of diameter, but, of course, the maximum error on the flow rate is larger because flow rate depends on diameter squared as well as pressure drop.

In handling the stated problem the experimental limitations were:

1. Rubber thickness varied from 0.040 to 0.5 in.
2. Outside diameter was 0.8 in. in each case.
3. Free inside diameter was 0.120 in.
4. Support plate hole diameter varied from 0.125 to 0.5 in.
5. Rubber hardnesses were approximately 35, 53 and 75 durometer.
6. Maximum pressure drop was 200 lb/sq in.
7. Maximum flow rate was 26 lb/in.
8. The range of T numbers was from 10,000 to 100,000.
9. Operating temperatures were from 70 to 95 F.
10. The fluid used was liquid water.

SUMMARY

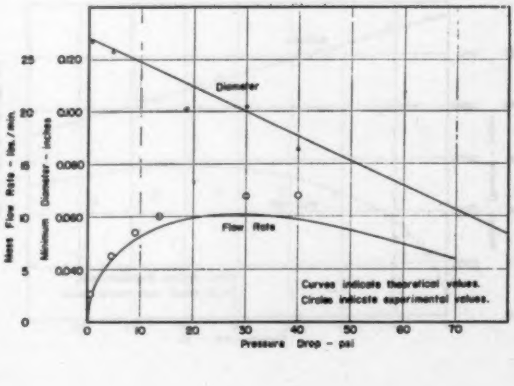
Restatement of the problem—Find a method which would allow for the prediction of the mass flow rate versus pressure drop relationship for given flow washer properties, back up plate geometry, and fluid at a specified state.

Limitation imposed by theory—The only flow washers considered were short, right circular cylinders with small cylindrical holes centered on the orifice axis. The back-up plates considered were flat with cylindrical, centered holes. The fluid was incompressible, flowing under steady state conditions.

Calculation of the mass flow rate—The steps to consider for the mass flow rate versus pressure drop calculations are:

1. The following information is assumed to be known: B. D, d′, t, durometer number, fluid density and viscosity.

Fig. 16 Comparison of theory and experiment for 53 durometer, 4/16 backup plate diameter, 0.500 thickness orifice



2. Consider Equations 1 and 22

$$M_m = \rho A \, KV' \qquad (26)$$

   The density ρ is constant and known; V' is given by Equation 20 as a function of ΔP and constants; A depends only on d which is a function of ΔP and geometry; and K may be obtained from ΔP and geometry.

3. It is recommended that d, K and V' be plotted as functions of ΔP and then the M_m versus ΔP relationship may be generated by use of Equations 1 and 22.

CONCLUSIONS

A method was developed for predicting the flow rate-pressure drop relationship for a given geometrical situation and rubber hardness for a flexible orifice. This method could be expanded to include a large variety of similar orifices, and it is possible that it could be extended into various geometries and orifice materials.

For thin orifices the flow coefficients were independent of Reynolds Number. For thick orifices the flow coefficient increased as Reynolds Number increased. The orifice may be considered circular when $\frac{\Delta d}{d'}$ is less than 0.4. Above this value the hole buckles.

In many cases the difference between the predicted and measured flow rates was less than 10% of the measured values. The maximum difference encountered was less than 25%. Due to the complexity of the problem it is felt that these comparisons are quite acceptable at this state in the solution of the problem of flexible orifices.

The region where the mass flow rate versus pressure drop relation had a negative slope was unstable. In each case there was either cavitation of the fluid or vibration of the orifice or both. It is not known which of the two phenomena (vibration, cavitation) is cause or effect.

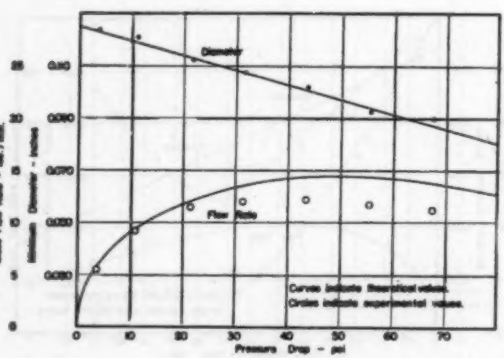


Fig. 17 Comparison of theory and experiment for 75 durometer, 7/16 backup plate diameter, 0.250 thickness orifice

BIBLIOGRAPHY

1. Holman, J. L. M., The Effect of a Variable Diameter Flow Orifice on the Performance of a Refrigeration System, Master Thesis, Purdue University, August 1957.
2. Timoshenko, S., Strength of Materials, Part II, Advanced Theory and Problems, Van Nostrand Co., 1941.
3. Rubber in Engineering, Chemical Publishing Co., Brooklyn, New York, 1946.
4. Burton, W. E., Engineering with Rubber, McGraw-Hill Book Co., 1949.
5. Handbook of Molded and Extruded Rubber, The Goodyear Tire and Rubber Co., 1949.
6. Elastomeric Engineering, Andre Rubber Company Ltd., England, 1945.
7. Roberts, I., Gaskets and Bolted Joints, ASME Journal of Applied Mechanics, Vol. 72, p. 169.
8. Binder, R. C., Direct Determinations of Coefficients for Flow Meters, Instruments, June 1942, p. 196.

DISCUSSION

CARL SPEICH, Columbus, Ohio: What were the maximum L/D ratios that were run on these tests? It is indicated that ratios up to about five were obtained. Did you go any further than this, and did L/D seem to have any bearing on stability?

AUTHOR ULRICH: The maximum thickness used was ½ in. The hole diameters were ⅛ in., so the ratio would be four. And then when the orifice diameter squeezes down to 50% of that, it would increase the L/D ratio to eight, so five is a good round number. This did have an influence on the stability. The thicker orifices and the softer ones were more unstable. The thinner ones tend to increase in diameter from the start. It is only the thicker ones that decreased in size which have an instability. The thicker ones would squeeze down until the hoop stress was enough to cause buckling. This was strictly unstable and it usually came from 40 to 60% decrease in the diameter.

HAROLD BLUM, Dallas, Tex.: How good is the assumption of uniform load; that is, uniform Δp, uniform load, when the rubber is bent or deformed? Is this a weakness, a serious weakness, in the approach theory?

AUTHOR ULRICH: Measurements were not made to check this pressure variation over the top surface of the orifice. On the largely twisted orifices, it probably would be a serious error. However, the angle of deflection for the large part, as far as the theory held, was in the order of 10 deg. So in this range, it is not serious. Also, the velocities were not especially high here. So until cavitation was obtained, which would come from flow around the corner, it was not considered serious. After cavitation, the theory did not hold. At that point, it probably became serious.

ERIK H. JENSEN, Staunton, Va.: Once cavitation occurred was there wear in the materials?

AUTHOR ULRICH: Extensive tests were not run on this and no cavitation wear on the rubber itself was noticed. It is not the same as metal, where cavitation might erode away the material. None was noticed.

1756

No. 1756

Degradation of Polyester* Film by Alcohols when Used as Additives in Refrigeration Systems

CLAUS J. BUSHOUSE

In March 1958 the author's company introduced a new insulation system for use in stators applied to hermetically sealed refrigeration and air conditioning systems. The new insulation system consists of (1) Acrylic coated magnet wire, (2) Polyester film sheet insulation and (3) Polyester covered lead cable and polyester lacing cord. We believe that this insulation system has superior properties, such as higher temperature capabilities, non-water forming, and the possibility of shorter, higher temperature dehydration. The properties of this system are described in greater detail by Hurtgen and Mounce.[1]

However, validation activity to determine the full extent of applicability of this new system has shown it to be vulnerable to methanol. The point of alcohol attack is the polyester film sheet insulation, chemically designated polyethylene terphthalate. Methanol at elevated temperatures degrades the film to the point of embrittlement. This has been established by sealed tube work and compressor testing. The preliminary work in sealed tubes gave results which indicated that methanol tends to embrittle the film. These reusable steel tubes are 14 in. long with a 1½-in. diam, have an approximate volume of 350 ml, and are shown in Fig. 1. The over-all length is approximately 17 in. and they can be taken apart readily at the "Ermeto fitting."

Each tube contained 35 gm Suniso 3G oil, ten gm polyester film, ten gm acrylic magnet wire and five gm aluminum casting sprue. Twelve of these tubes containing the above items were dehydrated and evacuated for four

* Mylar of E. I. duPont de Nemours & Co., Inc.

Claus J. Bushouse is active in the Hermetic Motor Department of the General Electric Company. This paper was prepared for presentation at the ASHRAE Annual Meeting, Denver, June 26-28, 1961.

hours at 150 C. After cooling, 24 mg of methanol were added to each of six tubes. To three of these, seven gm of Refrigerant 12 were added, and to the other three, five gm of Refrigerant 22. The remaining six tubes contained refrigerant with no methanol; three with seven gm of Refrigerant 12, and three with five gm of Refrigerant 22. Four tubes (one of each variety) could therefore be aged at each of three temperature levels. The condition of the film with and without methanol vapor after 30 days aging at 130, 140 or 150 C is given in Table I.

It is evident that the presence of methanol is linked with a more brittle film. Follow-up work using two ¼-hp freezer units also produced quite brittle film. Each unit contained 180 gm Refrigerant 22 plus 1 cc of methanol. One was run at a winding temperature of 130 C for 82 days and the other at 140 C for 60 days.

Since methanol reportedly is utilized to a limited extent in refrigeration units to prevent capillary freeze-up, it was necessary to investigate this situation and try to disclose some more favorable antifreeze compounds. First of all, it must be stated that the use of any antifreeze is not recommended for the acrylic-polyester insulation system. It is felt that there is no need since this system is non-hygroscopic and thus generates little or no water with little chance for a capillary tube freeze-up.

Investigation into the reactivity of possible antifreezes, therefore, included eight alcohols and two additional solvents, p-dioxane and tetrahydrofuran, both soluble in water and oil. Other antifreeze compounds such as glycols had too high boiling points and could sludge the system; aldehydes are prone to polymerize and ketones may halogenate. The best additive, therefore, had to be a good antifreeze, preferably less reactive than water, and should be compatible with all components of the system.

Table I

30 Day

Steel Sealed Tube Test

	Tube	Condition of Film
130 C	R-12	flexible
	R-12+MeOH	brittle at ¼ in. r
	R-22	flexible
	R-22+MeOH	brittle at ¼ in. r
140 C	R-12	brittle at ¼ in. r
	R-12+MeOH	brittle at ½ in. r
	R-22	flexible
	R-22+MeOH	brittle at ¼ in. r
150 C	R-12	brittle at ¼ in. r
	R-12+MeOH	brittle at ½ in. r
	R-22	just brittle (at 180° bend)
	R-22+MeOH	shatter brittle

PROCEDURE

The film was heated in a specially designed tube in the presence of each of these compounds, and also water whose hydrolytic reactivity was used as a sort of standard. The film also is sensitive to hydrolysis by water. However, this reaction is quite dependent upon the concentration and the temperature. At room temperature, hydrolysis of the film is measured in centuries, while at 200 C, in fractions of hours.

The glass apparatus designed

for this study is pictured in Fig. 2. It is fabricated from 1-in. and 7-mm Pyrex tubing and has an overall length of about 19 in. The self-contained, closed-end mercury manometer is capable of readings up to about 250 mm and the entire apparatus can be placed in a 20-in. oven to follow pressure changes through the glass observation panel. No transpiration corrections are necessary because both manometer and reaction tube are at the same reaction temperature. The alcohol in vapor form is added in sufficient quantity so that at reaction temperature the completely vaporized compound does not exceed the manometer limitations. The volume of the tubes used for the runs was relatively constant at 140 ml ± five ml so that it was necessary to follow pressure changes only.

The partially fabricated tube is cleaned thoroughly with detergent, rinsed with C. P. acetone, and dried at 105 C. After cooling, sufficient triple distilled mercury is added to the tube, which then is warmed slightly and evacuated to less than 20 micron using a clean rubber stopper as a closure. The tube is tilted and mercury is allowed to run into the manometer. Air is admitted slowly, the stopper removed, and 20 gm of ten-ml polyester film strips, 6 x ½ in., are dropped into the tube. The tube is then finished by sealing on a glass cap. The cap is a one-in. piece of pyrex tubing of the same diameter as the body of the reaction tube, to which a small stopcock has been sealed. The upper end of this stopcock has been sealed to a ground joint for attachment to the

Fig. 1 Reusable steel tube for sealed tube aging

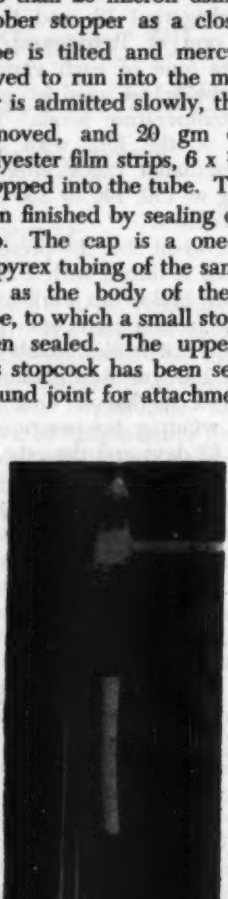
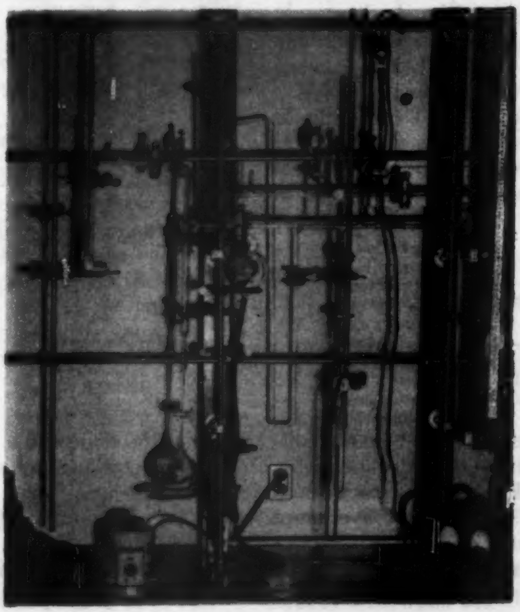


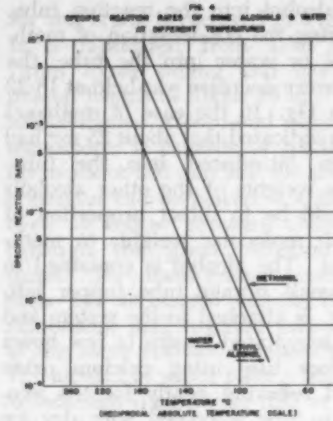
Fig. 2 Glass reaction tube with film and alcohol

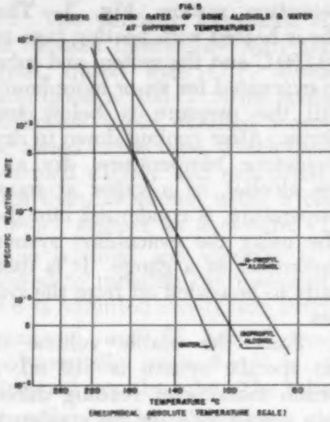
evacuation system, Fig. 3. The tube is heated with heating tape to 150-160 C and the system and tube are evacuated for six or more hours until the pressure is below ten micron. After cooling down to dry ice-acetone temperature, dry air-free alcohol, as a vapor at room temperature, is condensed into the tube using the evacuation system manometer as a gauge. It is then ready to be sealed off from the system.

Since the usable volume of this specific system is 810 ml, a certain manometer reading difference would indicate the condensation of a known suitable amount of alcohol into the reaction tube. During the condensation of methanol or water into the tube, the pressure decrease was held at 18-20 mm Hg. In the case of methanol this indicated that about 25 mg had been introduced into the tube. The weights of the other alcohols would be in direct proportion of their molecular weights to methanol. The alcohol is contained in a small storage tube (upper left, Fig. 3) attached to the system and is dehydrated in situ (a few hours before use) using calcium oxide and refluxing gently for ten min. This tube is cooled with dry ice and evacuated to remove air; as a

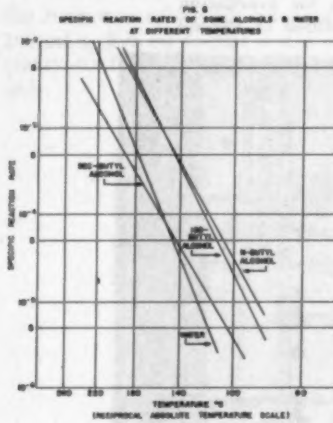
Fig. 3 Glass apparatus for evacuation of reaction tube and addition of alcohol

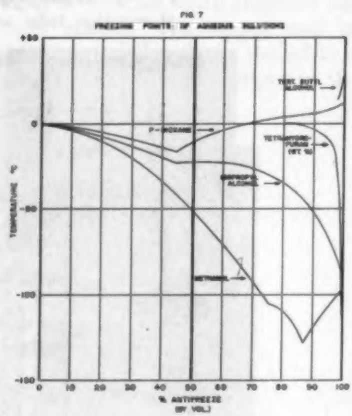






Figs. 4-6 Specific reaction rates per minute for some alcohols and water with polyester film. Fig. 7 Freezing points of aqueous solutions





further precaution, just before condensation into the reaction tube, the alcohol is allowed to expand into the system four or five times and evacuated each time.

The prepared tube is placed into an oven with an observation panel in the door and a steel millimeter scale is positioned between manometer and tube. Pressure readings are taken easily from the outside using flashlight and reading glass. Reading time intervals vary from five min to twice a day.

The higher temperatures obviously require more frequent measurements. The materials used were as follows:

Polyester film (polyethylene terphthalate) 1000A Type A.
Methanol, Baker Analyzed Reagent.
Ethyl alcohol, 190° grain Neutral Spirits (Publicker).
N-propyl alcohol, MCB 2852.
Isopropyl alcohol, Fisher Certified A-416.
N-butyl alcohol, Baker Analyzed Reagent.
Isobutyl alcohol, MCB 2858.
Sec-butyl alcohol, MCB 5047.
Tertiary butyl alcohol, Eastman 820.
Tetrahydrofuran, Baker Analyzed Reagent.
P-dioxane, Eastman 2144.

Although some of the above compounds were known to have properties causing them to be of doubtful use as antifreezes, the experiment was carried out with each one in order to make the study as complete as possible.

RESULTS

Pressure readings in millimeter of mercury were plotted on semi-log paper against linear time in minutes. The log pressure values follow a straight line and the reaction rate could readily be calculated for each temperature by using the following equation:

$$k_T = \frac{2.303}{t_2 - t_1} \times \log \frac{P_1}{P_2}$$

This is the integrated form of $\frac{-dc}{dt} = kc$ with pressure used in place of concentration when volume is constant

Where k_T = specific reaction rate
$t_2 - t_1$ = time interval during which pressure drops from P_1 to P_2.

The specific reaction rates for each reactive compound at three or more temperatures then were plotted on semi-log paper against the reciprocal of the absolute temperature. The position and slope of these lines give an indication of the relative reactivity of the alcohols studied; the smaller the number at a specific temperature, the less reactivity. For example, a rate of 1×10^{-4} or 0.0001 means that 0.01% of the alcohol is reacting with the film each minute. The Arrhenius equation $\frac{d\ln k}{dT} = \frac{E}{RT^2}$ whose integrated form is $\log k = \frac{E}{2.303 \times R} \left(\frac{1}{T}\right) + C$, is the basis for this plotting method, which shows that a straight line results from plotting log k against 1/T.

The experimental results are tabulated in Table II, and plotted in Figs. 4, 5 and 6. According to these results, the relative reactivity of the compounds studied is as follows:

1. No reaction
   - p-dioxane
   - tetrahydrofuran
   - tertiary butyl alcohol
2. Some reactivity
   - water
   - secondary alcohols—
     - isopropyl alcohol
     - secondary butyl alcohol
3. Good reactivity
   - primary alcohols—
     - methanol
     - ethyl alcohol
     - normal propyl alcohol
     - normal butyl alcohol
     - isobutyl alcohol

DISCUSSION

Reaction between the alcohols and

the film is termed "alcoholysis." In this instance it is a simple ester interchange. Polyethylene terphthalate has a molecular weight of about 15,000-16,000 and is a polymer consisting of about 75-80 units, each having a molecular weight of 192. Embrittlement of the film by an alcohol, methyl, for example, involves the rupture of at least one of the approximately 150 ester linkages in the chemical equation.

The chain has now been broken, in the simplest case, into two fragments; one fragment ending in the half ester of ethylene glycol and the other fragment in a methyl ester.

Reactivities of the alcohols with the polyester film fall into groups corresponding to the three alcohol classes; primary, secondary and tertiary. Primary alcohols were found to be the most reactive, ter-

$$\sim\sim\sim\; \overset{\overset{\displaystyle O}{\|}}{C}\text{-O-}\underset{H}{\overset{H}{C}}\text{-}\underset{H}{\overset{H}{C}}\text{-O} \;\vdots\; \text{-}\overset{\overset{\displaystyle O}{\|}}{C}\text{-}C_6H_4 \;\sim\sim\sim \;+\; CH_3OH \rightarrow$$

cleavage line

$$\sim\sim\sim\; \overset{\overset{\displaystyle O}{\|}}{C}\text{-O-}\underset{H}{\overset{H}{C}}\text{-}\underset{H}{\overset{H}{C}}OH \;+\; CH_3O\overset{\overset{\displaystyle O}{\|}}{C}\text{-}C_6H_4 \;\sim\sim\sim$$

Table II

Specific Reaction Rates With 10 Mil Polyester Film
(Multiply by 10^{-4} min^{-1})

	120 C	130 C	140 C	150 C	160 C	170 C	175 C	180 C	197 C	200 C
Water					1.44 1.48			5.29 5.54		19.6 21.3
Methanol	1.25 1.37	3.43	5.00 7.40		23.6 28.8					
Ethyl alcohol		1.19 (a) 1.97	3.36	5.45	10.4	20.0				
N-propyl alcohol .	1.31	1.81	2.91 4.26	6.13	10.9			32.4		
Isopropyl alcohol .	0.193		0.535 0.623	0.885	1.78 2.07	2.72				
N-butyl alcohol ..				6.31	13.3		25.2 31.2		63.7	90.5
Sec-butyl alcohol .					1.48	1.67		3.21		8.44
Isobutyl alcohol ..	1.02 0.856		3.78 4.16		12.3 12.8					
Tert-butyl alcohol	No reaction									
Tetrahydrofuran	No reaction									
P-dioxane	No reaction									

Note (a) = 136 C

tiary butyl alcohol had no reactivity and the two secondary alcohols were in between. The reactivities of the three alcohol classes follow very closely the ranking of reaction rates of esterification for these alcohols with acetic acid.[2]

On the basis of the above ranking according to the reactivity with the film, the compounds of choice would be those in the non-reactive group. These, however, are poor antifreezes as noted in Fig. 7. Furthermore, according to the results, the secondary alcohols have the same order of reactivity as the water whose freezing point is to be lowered. Secondary butyl alcohol has another disadvantage in that it is not completely soluble in water at all concentrations. According to some cursory work with isopropyl alcohol in the presence of aluminum foil, iron filings and film using the same type of reaction tube, there is evidence that the reactivity is greater at 160 C and above than without the metals. Strictly speaking, therefore, none of the compounds studied can be recommended for use as additives in a refrigeration unit having this polyester film as sheet insulation in the hermetic motor.

SUMMARY

Since the possibility of the use of methanol as a remedial measure in hermetic refrigeration units exists, various compounds were studied in order to determine their reactivity with polyethylene terphthalate film used in the new insulation system. Preliminary work showed that methanol is quite reactive, causing rapid embrittlement of the film. Studies of each alcohol with the film in a specially designed glass tube indicated that methanol and four other primary alcohols have the same order of reactivity; that two secondary alcohols were about equal to water in reactivity and that tertiary butyl alcohol had no reactivity. No recommendations can be made since the most likely substitute, isopropyl alcohol, is just as reactive as water; it might fall short in antifreeze properties, and it might cause an increase in non-condensable gases when used in the hermetic system. The conclusion, therefore, is that any of the ordinary antifreeze alcohols when added to the system would cause rapid degradation of the polyester film in this new insulation system.

REFERENCES

1. J. P. Hurtgen and A. R. Mounce, "New Insulation System for Hermetic Refrigeration Motors," ASHRAE JOURNAL, p 60, July 1959.
2. G. H. Richter, "Textbook of Organic Chemistry," 2nd. ed., p 191, Wiley, New York, 1943.

DISCUSSION

WALTER WALKER, Coral Gables, Fla.: It is indicated on the first page that there is no necessity of using antifreeze because these films do not give water to the system. However, there are many other ways in which water gets into the system other than just from the film. In the matter of testing, glycols were left out of the testing series, with the statement that they would sludge the system. Glycols have been used as antifreezes successfully in the past in systems and are still being used. There was no evidence of sludging difficulty when they were in use.

Trade mixtures of methanol and sodium methylate were not included in the tests; but these mixtures are added to machines by service men. The addition of methanol provides trouble, but the fact is that it is added.

Were mercury gauges used in all of these tubes, and if so, might not the mercury have catalyzed this reaction?

AUTHOR BUSHOUSE: Mercury was present in all of the reaction tube experiments, however its presence was not considered deleterious at all. No tests were made to determine whether it was present in the film at all or whether it catalyzed. It is not felt that it catalyzed.

The comment about ethylene glycol was made on the basis of automotive experience with antifreeze of that type in the cooling system of an automobile engine.

E. T. NEUBAUER, Sidney, Ohio: Before the war we specified antifreezes; since that time, however, we have not used them at all. With the system clean and dry and with an appropriate dryer, where necessary, it is possible to get along without antifreezes. The experiments with Mylar film reveal the necessity to get away from antifreezes. However, several organizations still specify the use of antifreezes.

W. WALKER: We have never recommended the use of methanol or other antifreezes. The practical question is that it is impossible to prevent the use of methanol and proprietary mixtures as antifreezes regardless of everything that is said against their use.

LYLE F. ALBRIGHT, Lafayette, Ind.: With regard to ethylene glycol, could it be assumed that since it is also a primary alcohol it might react similarly to the methanol and ethanol?

AUTHOR BUSHOUSE: It seems quite certain that it would. That is one of the starting points of the film.

1757

No. 1757

Solubility of Refrigerants 11, 21, and 22 in Organic Solvents Containing a Nitrogen Atom and in Mixtures of Liquids

ALLEN THIEME   LYLE F. ALBRIGHT

Absorption cooling cycles use a combination of absorption and stripping steps instead of the mechanical compressor of the vapor compression cycle. Advantages of the absorption cycle include lower operation costs if efficient absorption and stripping steps can be realized. Since the efficiency of these steps is determined largely by the chemical and physical properties of the solvent-refrigerant combination, evaluation of these properties is of importance in the development of an economical absorption cooling cycle.

From a consideration of the absorption cooling cycle, an efficient cycle would be one in which a large amount of refrigerant is evaporated in the generator from a given amount of solution. Such a solvent-refrigerant system would be one in which the refrigerant is soluble in the solvent in large excess of the amount predicted by Raoult's Law.

Using this supposition, Zellhoefer and co-workers[2,3,4,11,12] measured the solubilities of several halogenated methanes in many organic solvents. They found that halogenated methanes which contained a hydrogen atom in the molecule produced excess solubility in organic solvents in many cases. This excess solubility was attributed to "hydrogen bonding," which has since been shown to be essentially an electrostatic attraction between molecules.[7] Other investigators[1,8,9,10] have used this

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principle to seek a good solvent-refrigerant combination for the absorption cooling cycle. One of the best solvents found to date is the dimethyl ether of tetraethylene glycol. Eiseman[5] calculated coefficients of performance (C.O.P) of this solvent with several refrigerants and found a C.O.P. as high as 0.375.

In this investigation, the solubilities of three fluorochloro methanes were studied in several organic compounds containing a nitrogen atom. In addition, a preliminary attempt was made to evaluate binary mixtures of low volatile organic liquids as solvents.

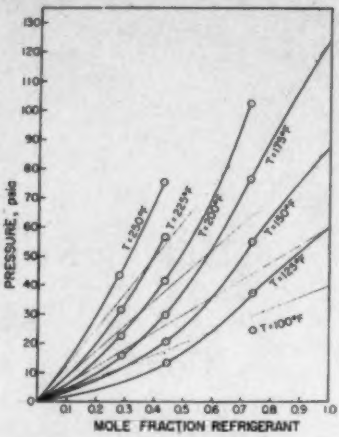


Fig. 1 Solubility of Refrigerant 21 in diethyl formamide

EQUIPMENT AND OPERATING PROCEDURE

Vapor pressure measurements were made using static methods. The apparatus utilized and the procedure used were essentially the same as those previously reported.[1]

Refrigerants used in this investigation were all commercial grade products.

Most of the solvents were also commercial grade materials.

RESULTS

Solubility data were obtained for a total of sixteen binary and nine ternary systems. Equilibrium pressures for each system were plotted

Table I Qualitative Results of Solubility Studies of Binary Mixtures

Solvent	Ref. 11 (CCl_3F)	Ref. 21 ($CHCl_2F$)	Ref. 22 ($CHClF_2$)
Tetraethylene Pentamine	Reacted	Reacted	Reacted
Dimethyl Formamide (DMF)	VLS	VHS	VHS
Diethyl Formamide (DEF)	—	VHS	VHS
Ethyl Formamide	—	LS	—
Dimethyl Aniline	—	HS	—
m-Chloro Aniline	—	LS	—
Aniline	—	LS	—
n-Octyl Amine	—	HS	—
n-Octyl Cyanide	—	HS	—
N-Methyl Morpholine	—	HS	—
Nitrobenzene	—	AS	—

VLS—Very low solubility
LS—Low solubility
AS—Average solubility
HS—High solubility
VHS—Very high solubility

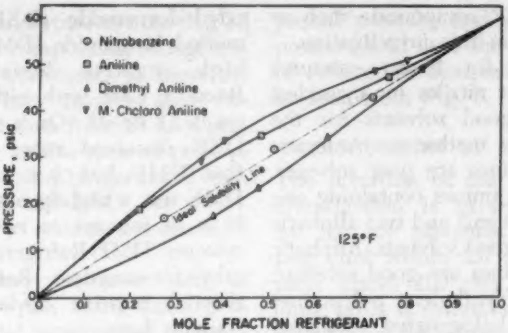
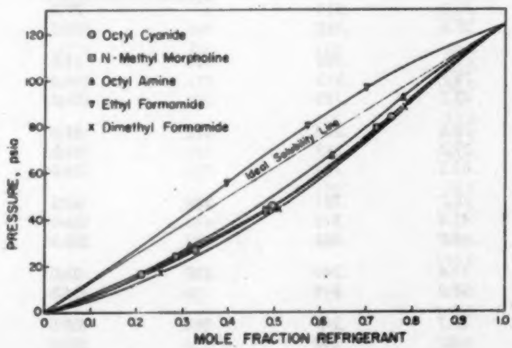


Fig. 2 Solubility of Refrigerant 21 in some closely related solvents at 125 F

versus the mole fraction of the refrigerant, with temperature as a parameter. Fig. 1 indicates the results for the system of Refrigerant 21 and diethyl formamide (DEF). In all cases, a system that showed either positive or negative deviations from Raoult's Law at one point showed similar deviations over all ranges of temperature and pressure. The qualitative comparison of the solubility results for the binary mixtures are given in Table I and Figs. 2 and 3 give quantitative comparisons for the solubility of Refrigerant 21 in aromatic solvents and in miscellaneous aliphatic solvents, respectively. Hydrogen bonding was found to be an important factor in

Fig. 3 Solubility of Refrigerant 21 in various solvents at 175 F



solubilities of compounds such as those used in this investigation.

Results for binary mixtures indicate that nitriles (or cyanides) are fairly good solvents for the halogenated methanes. Aromatic primary amines are poor solvents, but tertiary amines containing one aromatic nucleus and two aliphatic groups are good solvents. Aliphatic primary amines are good solvents; however, polyethylene polyamines react with halogenated methanes to form complex mixtures of water-soluble amine salts. Perhaps the aliphatic amine also may have reacted to a small but undetected extent.

The most promising solvents studied in this investigation were the di-N- substituted formamides and the basic solubility data obtained for these systems are shown in Tables II through V. Both diethyl formamide (DEF) and dimethyl formamide (DMF) showed high negative deviation from Raoult's Law with either Refrigerant 21 or 22. On a mole basis, DEF dissolved more refrigerant than DMF; but on a weight basis, DMF was a slightly better solvent. It is of interest to note that the systems DMF-Refrigerant 11 and ethyl formamide – Refrigerant 21 showed positive deviations from Raoult's Law.

Mixtures of two organic liquids also were evaluated as solvents for the halogenated methanes. Nine ternary systems were studied and these are listed in Table VI. All of these ternary systems except DMF – aniline – Refrigerant 21 showed negative deviations from Raoult's

Table II

Solubility of Refrigerant 21 in N,N-dimethylformamide

Temperature F	Pressure psia	Mole Fraction Refrigerant
100	4.4	.256
	13.2	.514
	28.5	.785
125	7.1	.255
	19.1	.513
	43.2	.785
150	10.8	.254
	29.7	.512
	63.3	.783
175	16.7	.251
	42.4	.511
	88.2	.782
200	23.6	.249
	59.2	.510
225	32.7	.247
	80.7	.508

Table III

Solubility of Refrigerant 22 in N,N-dimethylformamide

Temperature F	Pressure psia	Mole Fraction Refrigerant
100	19.5	.212
	56.0	.483
	96.0	.639
125	28.0	.209
	79.5	.480
	130.0	.627
150	39.0	.205
	111.0	.475
	171.0	.617
175	52.0	.201
	146.0	.470
	216.0	.595
200	67.5	.196
	186.5	.465
	265.5	.577
225	86.0	.189
	234.5	.458
250	107.5	.182
	288.0	.452

Law. It is important to note that all of the pure compounds, except aniline, that were used to form the binary solvents dissolved more Refrigerant 21 than Raoult's Law predicts. Pure aniline showed positive deviations with Refrigerant 21 and, when mixed with DMF, the resulting ternary system approached ideality.

The binary mixtures used as solvents for the refrigerants produced partial pressures of refrigerant above the ternary solution that are approximately a linear function of the solvent composition, at constant temperature and mole fraction refrigerant, as shown by Fig. 4. Three different mixtures of the dimethyl ether of tetraethylene glycol (DME-TEG) and DMF were evaluated as solvents in Fig. 4. The approximate straight line relationship between refrigerant partial pressure and solvent composition held over the entire range of refrigerant composition, as can be seen from Fig. 5.

DISCUSSION OF RESULTS

The precision of the data in this investigation is generally within 1.0 psi for pressures below 100 psig; and within 2.0 psi for pressures above 100 psig. The maximum error is estimated to be less than 2.5 psi for pressures up to 100 psig, and less than 5.0 psi for pressures above 100 psig.

A comparison has been made of the solvent capacities of DMF and of DEF with those of DME-TEG, which has been suggested as a good solvent for the halogenated methanes[5,6]. On a mole basis, DME-TEG is best. However, DME-TEG has a molecular weight

Table IV

Solubility of Refrigerant 21 in N,N-diethylformamide

Temperature F	Pressure psia	Mole Fraction Refrigerant
100	24.6	.738
125	13.1	.441
	37.2	.736
150	20.5	.439
	54.9	.733
175	15.8	.288
	29.7	.437
	76.2	.729
200	22.7	.286
	41.2	.434
	102.2	.725
225	31.7	.282
	56.3	.431
250	43.2	.278
	75.2	.427

Table V

Solubility of Refrigerant 22 in N,N-diethylformamide

Temperature F	Pressure psia	Mole Fraction Refrigerant
100	25.5	.324
	61.0	.533
	138.0	.785
125	36.5	.320
	85.5	.530
	188.0	.780
150	52.5	.315
	118.5	.525
	253.0	.773
175	69.5	.310
	157.5	.520
200	89.5	.304
	198.0	.516
225	113.5	.298
	248.0	.510
250	141.5	.289

Table VI Qualitative Solubility Results for Ternary Systems

Solvent	Refrigerant	Results
Dimethyl Formamide (DMF) and Triacetin		
Mole Ratio $\frac{\text{Triacetin}}{\text{DMF}} = 1.07$	21	HS
Dimethyl Ether of Tetraethylene Glycol (DME-TEG) and Triacetin		
Mole Ratio $\frac{\text{DME-TEG}}{\text{Triacetin}} = 1.03$	21	VHS
DME-TEG and Triacetin		
Mole Ratio $\frac{\text{Triacetin}}{\text{DME-TEG}} = 1.41$	22	VHS
DMF—Aniline		
Mole Ratio $\frac{\text{Aniline}}{\text{DMF}} = 0.99$	21	AS
DME-TEG and Tetraethylene Pentamine		
Mole Ratio $\frac{\text{DME-TEG}}{\text{Amine}} = 0.97$	21	VHS
DME-TEG and DMF		
Mole Ratio $\frac{\text{DME-TEG}}{\text{DMF}} = 0.82$	22	VHS
DME-TEG and DMF		
Mole Ratio $\frac{\text{DME-TEG}}{\text{DMF}} = 0.39$	21	VHS
DME-TEG and DMF		
Mole Ratio $\frac{\text{DME-TEG}}{\text{DMF}} = 1.03$	21	VHS
DME-TEG and DMF		
Mole Ratio $\frac{\text{DME-TEG}}{\text{DMF}} = 3.50$	21	VHS

of 222, whereas the molecular weights of DMF and DEF are 73 and 101, respectively. When the solvents are compared on a weight basis as shown in Table VII and Fig. 6, DMF and DEF are better solvents than DME-TEG.

Eiseman[5] compared the efficiency of solvent-refrigerant combinations by calculating the coefficient of performance (C.O.P), pumping requirements, etc., with DME-TEG as a solvent, and various fluoro-chloro alkanes as refrig-

Table VII Comparison of the Solubility of Refrigerant 21 in Three Solvents at Atmospheric Pressure

	DMF		DEF		DME-TEG	
Temp. F	Moles/ Mole	Grams/ Gram	Moles/ Mole	Grams/ Gram	Moles/ Mole	Grams/ Gram
125	.779	1.10	.870	.885	1.35	.625
150	.463	.652	.558	.567	.840	.338
175	.292	.411	.355	.361	.504	.233
200	.191	.269	.261	.266	.322	.149

erants. Exactly the same step-by-step calculations are made in Table VIII, using different solvents, but two of the same refrigerants. In order to make the calculations in Table VIII, the specific gravities and specific heats of DMF and DEF were needed. The values used and the sources are listed.

DMF, specific gravity = 0.945 from Perry, J. H., "Chemical Engineers' Handbook" 3rd Edition, McGraw-Hill Book Co. (1950).

DEF, specific gravity = 0.908 from Lange, N. A., "Handbook of Chemistry", 8th Edition, Handbook Publishers, Inc. (1952).

DMF, specific heat = 0.5 from "DMF Product Information," Grasselli Chemicals Depts., E. I. duPont de Nemours and Co.

DEF, specific heat = 0.5, was estimated, based on the value for DMF.

The systems calculated in Table VIII are Refrigerant 21 with DMF and DEF (Columns 3 and 5), and Refrigerant 22 with DMF and DEF (Columns 4 and 6). The results are compared to the sys-

Fig. 4 Vapor pressures of Refrigerant 21 over solutions of dimethyl ether of tetraethylene glycol, dimethyl formamide and Refrigerant 21 as a function of solvent composition

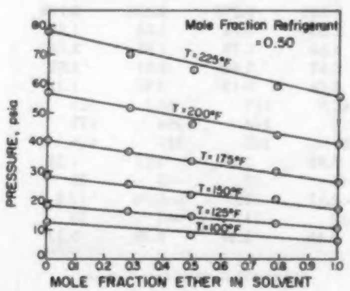


Fig. 5 Solubility of Refrigerant 21 in various mixtures of dimethyl ether of tetraethylene glycol and dimethyl formamide at 175 F

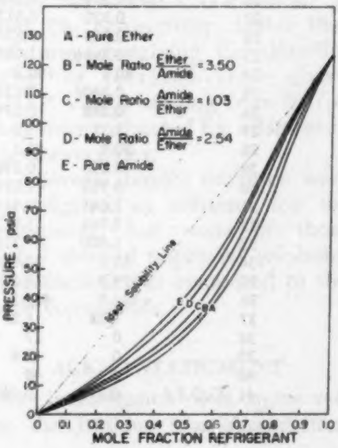


Table VIII Calculation of Performance Characteristics of Solvent —Refrigerant Combinations System

Step	DME-TEG + 21	DME-TEG + 22	DMF + 21	DMF + 22	DEF + 21	DEF + 22
Molecular Weight	102.9	86.5	102.9	86.5	102.9	86.5
Boiling Point, C ..	+8.9	—40.8	+8.9	—40.8	+8.9	—40.8
Boiling Point, F ..	+48.1	—41.4	+48.1	—41.4	+48.1	—41.4
1	105.2	87.4	105.2	87.4	105.2	87.4
2	10,820	7,560	10,820	7,560	10,820	7,560
Absorber						
Op. Temp., F ...	95	95	95	95	95	95
Op. Press, psia ..	12.3	83.7	12.3	83.7	12.3	83.7
3	0.680	0.748	0.52	0.62	0.57	0.65
4	49.6	53.6	60	66	57	62
5	2.125	2.97	1.08	1.63	1.33	1.86
Generator						
Op. Temp., F ...	232	232	232	232	232	232
Op. Press., psia ..	42.9	224.6	42.9	224.6	42.9	224.6
6	0.414	0.516	0.29	0.41	0.34	0.45
7	24.7	29.3	36	45	34	41
8	0.706	1.07	0.41	0.69	0.52	0.82
9	1.419	1.90	0.67	0.94	0.81	1.04
10	42.9	224.6	42.9	224.6	42.9	224.6
11	12.3	83.7	12.3	83.7	12.3	83.7
12	30.6	140.9	30.6	140.9	30.6	140.9
13	15,360	14,364	7,250	7,200	8,760	7,860
14	2.250	2.995	2.250	2.[illegible]5	2.250	2.995
15	3.425	4.051	2.40	2.67	2.77	3.20
16	4.567	5.401	3.20	3.56	3.70	4.27
17	1.494	2.238	1.80	2.91	1.91	2.97
18	3.744	5.233	4.05	5.91	4.16	5.97
19	8.311	10.634	7.25	9.47	7.86	10.24
20	1.379	1.213	1.379	1.213	1.379	1.213
21	0.325	0.517	0.352	0.584	0.362	0.590
22	0.540	0.639	0.407	0.451	0.488	0.564
23	0.8656	1.156	0.759	1.035	0.850	1.154
24	80.6	190.9	80.6	190.9	80.6	190.9
25	0.0408	0.129	0.0357	0.115	0.0400	0.129
26	0.088	0.279	0.077	0.249	0.087	0.279
27	3.73	11.8	3.28	10.5	3.67	11.8
28	20.9	66.1	18.7	58.9	20.5	66.1
29	0.255	0.298	0.255	0.298	0.255	0.298
30	0.955	1.559	1.03	1.76	1.06	1.78
31	2.041	2.414	1.60	1.78	1.85	2.14
32	2.996	3.973	2.63	3.54	2.91	3.92
33	1.000	1.326	0.88	1.18	0.97	1.31
34	52.2	91.4	62.9	119	66.7	121
35	279.7	330.7	219	244	254	293
36	331.9	422.1	282	365	321	414
37	1.00	1.27	0.85	1.10	0.97	1.25
38	0	27	—15	10	—3	25
39	0	12.8	—0.62	10.7	—0.11	12.8
40	0	40	—15	21	—3	38
41 (C.O.P.)	0.375	0.268	0.44	0.31	0.39	0.27

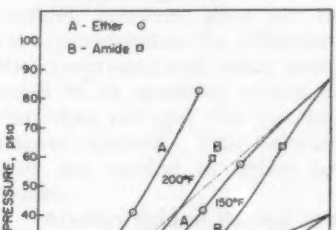
tems DME-TEG with Refrigerants 21 and 22 as calculated by Eiseman (Columns 1 and 2). Using DMF instead of DME-TEG as the solvent increases the C.O.P. by about 15%, DEF results in a 1-3% increase in the C.O.P., as compared to DME-TEG.

Although DMF and DEF are good solvents based on solubility considerations, both have relatively low boiling points, so that they might cause problems in an absorption refrigeration system. In this respect, N,N-dimethylacetamide has a considerably lower volatility and this compound should be tested in the future. The N,N-dialkylformamides may also require additional investigation to determine if they are adequately stable in a refrigeration cycle.

Although the ternary solutions used in this investigation did not increase solubility, they may prove useful in improving solvent properties such as viscosity, stability and corrosiveness. For instance, tetraethylene pentamine reacted with the refrigerant, but a 50:50 mixture of DME-TEG and tetraethylene pentamine did not appear to react.

In future work on ternary systems, it would be of interest to study the effects of adding a highly associated liquid or a polar salt to an organic solvent. Such additions might increase the solubility of the refrigerant in the solution.

Fig. 6 Comparison of dimethyl ether of tetraethylene glycol and dimethyl formamide on a weight basis



CONCLUSIONS

Several nitrogen-containing organic solvents were tested; some amines, nitriles and formamides were found to be good solvents for Refrigerants 21 and 22, which contain a hydrogen atom and hence can hydrogen bond. The two best solvents found were dimethyl formamide and diethyl formamide. These two solvents produce mixtures that have C.O.P. about 15 and 2%, respectively, better than mixtures containing the dimethyl ether of tetraethylene glycol. C.O.P. values ranging from 0.27 to 0.44 were estimated for mixtures of the formamides.

Several binary mixtures were investigated as solvents for the refrigerant, but none of those tested showed improved solubility characteristics as compared to the pure compounds.

ACKNOWLEDGMENT

This investigation was sponsored by the Indiana Gas Association.

Details and the unabriged data can be found in Allen Thieme's M. S. thesis, Purdue University (1960).

REFERENCES

1. Albright, L. F., Buclez, P. C., Pluche, C. R., and Doody, T. C., ASHRAE Transactions, 66, 423-433 (1960).
2. Copley, M. J., Zellhoefer, G. F., and Marvel, C. S., J. AM. CHEM. SOC., 60, 2666-73 (1938)
3. Copley, M. J., Zellhoefer, G. F., and Marvel, C. S., J. AM. CHEM SOC., 61, 3550-2 (1939).
4. Copley, M. J. Zellhoefer, G. F., and Marvel, C. S., J. AM. CHEM. SOC., 62, 227-8 (1940)
5. Eiseman, B. J., ASHRAE JOURNAL, 1 (No. 12), 45, (1959).
6. Ellington, R. T., Kunst, G., Peck, R. E., and Reed, J. F., "The Absorption Cooling Process," Institute of Gas Technology, Research Bulletin No. 14, August 1957.
7. Hildebrand, J. H., and Scott, R. L., "The Solubility of Nonelectrolytes", 3rd Ed., Reinhold Publishing Corp., New York, 1950.
8. Mastrangelo, S. V. R., J. PHYS. CHEM., 64, 608 (1959).
9. Mastrangelo, S. V. R., ASHRAE JOURNAL, 1 (No. 10), 64 (1959).
10. Servel, Inc., Personal Communication 1958).
11. Zellhoefer, G. F., IND. ENG. CHEM., 29, 548-51 (1937).
12. Zellhoefer, G. F., Copley, M. J., and Marvel, C. S., J. AM. CHEM. SOC., 60, 1337-43 (1938)

DISCUSSION

H. O. SPAUSCHUS, Louisville, Ky.: Have you considered interpreting the results found in terms of a molecular structure of the nonvolatile component? Were any clues seen in terms of the kind of molecules that would be needed to contribute to high solubilities in what was being investigated beyond the hydrogen bonding aspect that was mentioned?

AUTHOR ALBRIGHT: Considerable thought has been given to interpreting the results in terms of molecular structure. A paper on this subject is being prepared for publication.

Concerning the types of solvent required to obtain considerably higher solubilities, organic solvents will probably not be suitable. Some atomic group is presumably needed that will give a true reversible (at high temperatures) chemical reaction. With refrigerants of the type studied here, it is doubtful if such a solvent can be found. Some amines were studied and a chemical reaction occurred; however, it was not reversible. Perhaps mixtures of solvents offer the best hope of obtaining high solubilities with chlorofluoromethanes; even this approach does not seem too promising, though.

1758

No. 1758

Refrigerating Capacity and Performance Data for Various Refrigerants, Azeotropes and Mixtures

R. C. McHARNESS D. D. CHAPMAN

Engineers frequently are interested in the relative cooling capacities of various compounds and azeotropes. Tables of thermodynamic properties or pressure-enthalpy diagrams can be used to calculate theoretical values for the individual refrigerants and compare data at any given set of operating conditions. This method, however, does not take into consideration the behavior of the compressor and other components of an operating system and thus does not give the "practical" answer desired. Unit operating tests are needed to obtain such values.

Another field of interest to refrigerating engineers is that of mixed refrigerants. Much of the work carried out thus far has been concerned with the increased capacity obtainable with mixtures and the heat-transfer problems and advantages involved in their use.[1,2,3] Our interest has been to expand the information available on azeotropes and mixtures and to compare their refrigerating capacity and operating data with similar data for pure compounds.

A calorimeter test unit has been built and used to attain the objectives mentioned previously. Although the cooling capacity values constitute the primary data developed in our unit, its instrumentation is such that a variety of other operating information is also available. Thus, power use values, the coefficient of performance, suction and discharge temperatures, suc-

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tion and discharge pressures, and the compression ratio also were obtained and are reported here.

Two compressors of the same model were used in this work. Since the studies covered a wide pressure range, there were significant variations in the volumetric efficiencies. These are reflected in the refrigerating capacities reported. While the values are entirely realistic for the conditions used, it is possible that capacity improvements can be obtained if the compressor characteristics are matched to those of the refrigerant and to the operating conditions desired.

CALORIMETER TEST UNIT

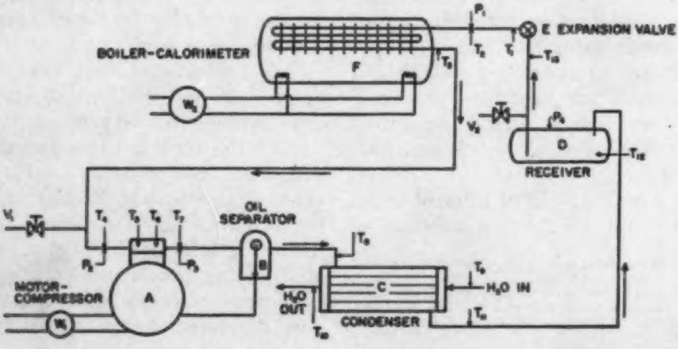
The test stand used in this work contains the basic components found in any refrigerating unit. A schematic diagram of the equipment is shown in Fig. 1. The points at which pressure and temperature are controlled or measured also are shown, along with the refrigerant sampling points.

This unit is similar to those used by many equipment manufacturers. At least one such unit has been described previously.[4] The various pieces of equipment, most of which are commercially available, are described briefly below.

A. **Motor-compressor—This semihermetic unit is made up of a 1.5-hp, 1750-rpm, 230-volt, 60-cycle, single-phase motor and a 2-cyl compressor having a calculated displacement of 169 cfh. An indicating wattmeter (W_1) is mounted in the power supply to the motor. A constant-voltage regulator is located in the power line serving both the motor and the boiler-calorimeter (F) heaters. This was found to be essential in attaining equilibrium conditions and making reproducible runs.**

The compressor is equipped with pressure gages and thermocouples on the suction line (P_2 and T_4) and the discharge line (P_3 and T_7) near the respective service valves of the compressor. A sampling valve (V_1) is located on the suction line. The compressor head is drilled and tapped to

Fig. 1 Schematic diagram of calorimeter test unit



take four thermocouples* (shown as T_5 and T_6). They are located in the suction and discharge gas streams of both cylinders at a point 1/16 in. from the valve reeds.

Actually two identical compressors, A and B, were used in this work. At the —20 F evaporator condition the two gave significantly different refrigerating capacities as shown by the data for Refrigerants 12 and 22 in Table V. In making comparisons at this temperature, it is necessary to use the data for the proper compressor. At evaporator temperatures of —10 and 40 F the results obtained with the two compressors were less than 2% apart. This is well within the accuracy of our measurements. Hence, the refrigerating capacities obtained with Compressor A were used in all comparisons made at the higher evaporator temperatures.

B. Oil Separator—A standard 1-hp Refrigerant 22 unit is used. It is equipped with screens on the inlet and outlet lines and with a float valve to control the return of oil to the compressor.

C. Condenser—The condenser is a tube-in-tube, water-cooled, counterflow unit made up of copper tubing with brass headers. The refrigerant flow is in the outer tubes. The water flow is controlled by a hand-operated valve. Thermocouples are located on the refrigerant inlet and outlet lines (T_8 and T_{11}) and on the water inlet and outlet lines (T_9 and T_{10}).

D. Receiver—A stainless steel vessel approximately 4.5 in. OD x 17 in. long with a volume of 180 cu in. is used as the receiver. It is wrapped with 0.5-in. OD copper tubing and insulated with polyurethane foam. The receiver is equipped with a liquid-level gage glass, a pressure gage (P_4), and a well in which a thermocouple (T_{12}) is located. Discharge water from the condenser (C) is circulated through the receiver coil. A sampling valve (V_3) is located on the liquid line which connects to a standpipe in the receiver.

E. Expansion Valve—An air-operated diaphram-type control valve is utilized. The air pressure and subsequent valve opening are controlled by the system pressure (P_1) downstream from the valve. Maximum sensitivity and control of refrigerant flow are obtained by the use of a valve positioner along with TFE-fluorocarbon resin rings in the packing gland to reduce starting friction to a minimum. The valve and its connecting lines are well insulated.

The temperature of the liquid going to the expansion valve is measured with a thermocouple (T_{13}) on the line. A similar thermocouple (T_1) is mounted on the line downstream from the valve. A calibrated mercury thermometer (T_3) is located in a well in the line below T_1. A pressure gage of high accuracy can be connected at the same point (P_1) or to the suction line (P_3) of the compressor.

F. Boiler - Calorimeter (Evaporator) —This is a stainless steel vessel approximately 13 in. OD x 32 in. long and having a total volume of about 3400 cu in. An evaporator of ample area is mounted in the upper part of the vessel while four 1000-watt heaters are located near the bottom. A complete range of heat inputs is obtained by using a variable voltage controller with one of the heaters. The total heat input over a fixed period of time is measured with a recording watthour counter (W_1). A 3.5-in. depth of Refrigerant 11 is used as the boiling heat transfer liquid in the boiler-calorimeter.

The vessel and its connecting lines are insulated thoroughly with a mininum of two in. of polyurethane foam. A thermocouple (T_2) is located on the outlet refrigerant line of the evaporator.

Complete evaporation of the refrigerant takes place in the evaporator. In addition, the vapor is superheated to 47-60 F at the —20 F evaporating temperature, to 55-60 F at —10 F, and to 62-63 F at 40 F. These temperatures are those measured at T_2. The superheat temperatures with Refrigerant 13B1 and its mixtures are always higher than those with the other refrigerants.

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* Conax thermocouples

Table I-A—Performance Characteristics of Various Refrigerants

Nominal Conditions: —20 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

	Data for Indicated Refrigerant								
Refrigerant	12		500[c]	13/12 (11/89 wt %)	115	22		502[d]	13/22 (10/90 wt %)
Compressor Used	A	B	A	B	B	A	B	A	B
Refrigerating Capacity									
Measured (Btu/hr)	1,250	1,150	1,525	1,470	1,675	2,360	2,175	2,650	2,500
% of Refrigerant 12	100	100	122	128	146	189	189	212	217
% of Refrigerant 22	53	53	65	68	77	100	100	112	115
Coefficient of Performance									
Actual	0.70	0.58	0.72	0.66	0.74	0.97	0.84	0.95	0.87
Indicated[a]	1.66	1.20	1.40	1.21	1.35	1.67	1.38	1.49	1.34
Theoretical	2.50	2.50	2.46	—	1.93	2.30	2.30	—	—
Compression Ratio	9.88	9.88	10.07	10.75	9.15	9.74	9.74	8.88	9.92
Suction Pressure (psia at P_2)	15.3	15.3	17.8	19.6	22.8	25.0	25.0	29.8	29.6
Discharge Pressure (psia at P_3)	151	151	180	211	209	243	243	265	294
Suction Temperature (F at T_5)[b]	134	131	130	129	102	142	141	116	136
Discharge Temperature (F at T_6)[b]	236	233	238	244	194	282	282	238	286
Motor Input (watt)	520	580	620	655	665	715	760	820	845
Evaporator Input (watt)	366	337	447	430	491	692	637	776	732

[a] Indicated C.O.P. = $\dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

[c] Azeotrope of Refrigerants 12/152a (73.8/26.2 wt %) [d] Azeotrope of Refrigerants 22/115 (48.8/51.2 wt %)

Table I-B—Performance Characteristics of Various Refrigerants

Nominal Conditions: —10 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Equivalent results are obtained with either Compressor A or B

	Data for Indicated Refrigerant						
Refrigerant	12	500	13/12 (11/89 wt %)	115	22	502	13/22 (10/90 wt %)
Refrigerating Capacity							
Measured (Btu/hr)	1,950	2,375	2,340	2,505	3,455	3,780	3,755
% of Refrigerant 12	100	122	120	128	177	194	193
% of Refrigerant 22	56	69	68	73	100	110	109
Coefficient Performance							
Actual	0.92	1.00	0.93	0.98	1.23	1.20	1.15
Indicated[a]	1.76	1.77	1.56	1.63	1.94	1.77	1.67
Theoretical	2.82	2.75	—	2.49	2.61	—	—
Compression Ratio	7.90	7.99	8.47	7.34	7.78	7.17	7.94
Suction Pressure (psia)	19.2	22.5	25.7	28.4	31.3	36.9	37.0
Discharge Pressure (psia)	151	180	218	209	243	265	294
Suction Temperature (F)[b]	125	119	118	96	128	105	126
Discharge Temperature (F)[b]	233	231	239	187	272	226	275
Motor Input (watt)	625	693	740	750	823	925	960
Evaporator Input (watt)	572	696	685	737	1,013	1,108	1,100

[a] $\text{Indicated C.O.P.} = \dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions} - \text{Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

Table I-C—Performance Characteristics of Various Refrigerants

Nominal Conditions: 40 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Equivalent results are obtained with either Compressor A or B

	Data for Indicated Refrigerant						
Refrigerant	12	500	13/12 (11/89 wt %)	115	22	502	13/22 (10/90 wt %)
Refrigerating Capacity							
Measured (Btu/hr)	8,660	10,155	10,620	10,040	13,660[c]	14,090[c]	7,400[d]
% of Refrigerant 12	100	117	123	116	158	163	—
% of Refrigerant 22	63	74	78	73	100	103	123
Coefficient Performance							
Actual	2.86	3.08	2.88	2.72	3.28	3.13	2.46
Indicated[a]	4.34	4.47	3.99	3.77	4.35	4.05	3.74
Theoretical	5.91	5.80	—	4.83	5.64	—	—
Compression Ratio	2.93	2.91	3.06	2.79	2.90	2.77	2.87
Suction Pressure (psia)	51.7	61.7	71.2	74.7	84.0	95.7	102.2
Discharge Pressure (psia)	151	180	218	209	243	265	294
Suction Temperature (F)[b]	86	84	82	75	86	79	102
Discharge Temperature (F)[b]	170	168	174	139	191	162	201
Motor Input (watt)	885	965	1,080	1,090	1,220	1,320	880
Evaporator Input (watt)	2,537	2,975	3,111	2,942	4,003	4,129	2,168

[a] Indicated C.O.P. $= \dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

[c] Voltage increased from 230 to 235 volt. This was necessary to obtain sufficient heater capacity at the evaporator. Compressor motor also operated at the higher voltage.

[d] Compressor operating on only 1 cyl.

G. Safety—Pressure-relief devices are provided on the test unit to prevent build-up of excessive pressure. Each consists of a rupture disc in a suitable mounting and each is set to burst at 440 psig. They are located on the suction and discharge lines of the motor-compressor, the inlet line to the condenser, the receiver and the boiler-calorimeter.

OPERATION OF THE CALORIMETER (See Fig. 1)

A. Evacuation—Prior to charging refrigerant to the calorimeter, extreme care is exercised in evacuating the entire system. This is done by means of a good vacuum pump to a pressure of 50 micron (0.05 mm of mercury) or less. Evacuation is continued until this low pressure has been maintained for 2 hours. Not only does this assure a system free of air and moisture, but it is also an excellent check on whether the system is free of leaks. Experience has shown that a pressure of 50 micron can not be reached with our pump if there is a leak in the system. The vacuum is applied at the sampling valves on the suction line to the compressor and the liquid line from the receiver (V_1 and V_2).

B. Charging—After the system and the loading line are evacuated to the required level, the unit is charged with the refrigerant to be tested. This is done from the liquid phase of the supply cylinder until the sight glass shows the receiver (D) is approximately half full. In this unit, the charge varies from 5 to 8 lb depending on the refrigerant being used. This procedure is followed regardless of whether pure compounds or mixtures are to be tested.

Mixtures are prepared by adding the exact desired weight of the higher-boiling component to a dry evacuated cylinder and then adding the exact required weight of the lower-boiling component. As described previously, a vacuum of 50 micron is pulled on the empty supply cylinder and also on the connecting lines before any refrigerant flow is permitted. The components actually are weighed into the supply cylinder through a capillary tube. This has been found to provide good control and ease of weighing for the preparation of mixtures to an accuracy of 0.05%. In order to assure uniformity of composition the cylinder is rolled prior to use.

A small cylinder is evacuated and filled slowly to a pressure well below saturation with vapor from the liquid phase of the supply cylinder. This is used as a standard in comparing the compositions of samples taken from the calorimeter with that of the material charged. The analyses are made with a gas chromatograph.

C. Attaining Equilibrium—A state of equilibrium must be reached in the system before any data are taken during a run. The conditions controlled to obtain and maintain this equilibrium are as follows:
1. The suction pressure at P_1 corresponding to the desired evaporating temperature.
2. The discharge gas pressure at P_2 corresponding to the desired condensing temperature (usually 110 F).
3. The suction gas temperature at T_4 (usually 65 F).

In the case of pure compounds for which tables of thermodynamic properties are available, the values for pressures P_1 and P_2 at the respective desired temperatures are obtained directly from the tables. The required value is established and maintained at P_1 by adjusting the flow through the expansion valve (E). The desired pressure is obtained at P_2 by controlling the flow of cooling water to the condenser (C). The third variable, the suction gas temperature at T_4, is controlled by varying the heat input to the boiler-calorimeter (F).

In the case of refrigerant mixtures or compounds for which tables are not available, suction and discharge gas control pressures must first be determined. This is done in the following manner. With the system operating, the expansion valve (E) is adjusted to obtain the desired evap-

orating temperature at T_3. The corresponding pressure is read at P_1. This pressure value then is established as the suction gas pressure at P_3 by adjustment of the expansion valve. Similarly, by manipulating the water flow to the condenser (C) the desired condensing temperature is established at T_{12} in the receiver (D) and the corresponding pressure read at P_4. This value then is used as the gas discharge pressure at P_2 for the run by further adjustment of the condenser water control valve.

D. Data Taking—With the attainment of equilibrium conditions, as evidenced by the constancy of the control values, P_3, P_2 and T_4, the watthour counter (W_2) on the boiler-calorimeter and a timer are activated. Approximately 20-30 min later, the counter and timer readings are recorded together with those of the indicating wattmeter (W_1) the motor-compressor (A) and the various pressure and temperature points. This information is made a part of the permanent record of each calorimeter test run. The same procedure is repeated during the period of the run, approximately 3 hr, with readings being made each 20 to 30 min until 3 or more identical sets of readings are obtained.

E. Sampling—Upon completion of the data-taking and while the unit is still at equilibrium conditions, a sample of the refrigerant vapor is withdrawn into a small evacuated cylinder from the sampling valve (V_1) in the suction line. The composition of this vapor sample is then compared with that of the standard taken from the supply cylinder prior to the start of the test. In all cases the gas being circulated had the same composi-

Table 11-A—Performance Characteristics of Refrigerants 22/12 Mixtures

Nominal Conditions: —20 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor A

	Data for Indicated Refrigerants 22/12 Mixtures (wt %)					
	0/100	25/75	50/50	75/25	85/15	100/0
Refrigerating Capacity						
Measured (Btu/hr)	1,250	1,625	2,135	2,315	2,355	2,360
% of Refrigerant 12	100	130	171	185	189	189
% of Refrigerant 22	53	69	90	97	100	100
Coefficient of Performance						
Actual	0.70	0.78	0.92	0.96	0.97	0.97
Indicated[a]	1.66	1.54	1.64	1.67	1.66	1.67
Theoretical	2.50	—	—	—	—	2.30
Compression Ratio	9.88	9.54	9.40	9.46	9.51	9.74
Suction Pressure (psia)	15.3	20.4	23.5	24.8	25.1	25.0
Discharge Pressure (psia)	151	195	221	235	239	243
Suction Temperature (F)[b]	134	130	134	137	137	142
Discharge Temperature (F)[b]	236	249	262	273	275	282
Motor Input (watt)	520	610	680	705	715	715
Evaporator Input (watt)	366	476	625	678	690	692

[a] Indicated C.O.P. = $\dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

tion as that of the refrigerant charged. The gas chromatograph used in making these analyses gave composition values with an accuracy of 0.1% or better.

The liquid refrigerant being circulated was analyzed in a few runs. The sample was taken at valve (V_2) and the procedure followed was that used in obtaining the standard sample from the supply cylinder (See Sect B).

RESULTS AND DISCUSSION

A. Test Conditions – Most of the tests were carried out at nominal evaporator temperatures of –20, –10 and 40 F and a nominal condenser temperature of 110 F, with the suction gas superheated to 65 F. These conditions were chosen as representative of those that would be expected in many refrigerating and air-conditioning applications. In addition, a few runs were made at nominal evaporator temperatures as low as –60 F.

In making our measurements, the suction and discharge pressures equivalent to the desired evaporator and condenser saturation temperatures (–20 and 110 F, for example) were fixed at points close to the compressor (P_2 and P_3). Hence, the evaporator and condenser temperatures reported are nominal values. The true evaporator temperatures are 2-5 F higher and the true condenser temperatures are 0-2 F lower than the nominal. These differences are due to the pressure drops in the respective portions of the system.

B. Data Reported–The manner of reporting the results and the bases on which they are derived are the same in all the tables. Thus, the measured refrigerating capacities (see Table I-A) are calculated from the evaporator power inputs reported, on the basis of a 1-hr time period. Watthours are converted to Btu/hr by multiplying by the factor 3.413.

Increased cooling capacity requires increased power input to the motor. This is shown by the values reported for motor and evaporator (power) inputs.

The actual C.O.P. values reported are obtained by dividing the evaporator input by the motor input. In reporting the indicated C.O.P.s the power required by the motor-compressor under no-load conditions with the suction valves removed (300 watt) is subtracted from the total motor input.

The theoretical C.O.P. values are calculated from the tables of thermodynamic properties using the nominal operating conditions specified. Each value is the ratio of the net refrigerating effect to the heat of compression at constant entropy. The net refrigerating effect is obtained by subtracting the enthalpy of the liquid at the condensing temperature (usually 110 F) from the enthalpy of the superheated vapor entering the compressor (at the suction pressure (P_2) and usually at 65 F). The heat of compression is obtained by subtracting the above enthalpy of the superheated vapor entering the compressor from the enthalpy of the vapor after compression along an isentrope to saturation pressure (P_3) at the nominal condenser temperature (usually 110 F).

Suction and discharge pres-

Table II-B—Performance Characteristics of Refrigerants 22/12 Mixtures

Nominal Conditions: —10 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor A

	Data for Indicated Refrigerants 22/12 Mixtures (wt %)						
	0/100	12/88	25/75	50/50	75/25	85/15	100/0
Refrigerating Capacity							
Measured (Btu/hr)	1,950	2,255	2,665	3,195	3,380	3,470	3,455
% of Refrigerant 12	100	115	137	164	173	178	177
% of Refrigerant 22	56	64	77	92	98	100	100
Coefficient of Performance							
Actual	0.92	0.97	1.12	1.20	1.22	1.24	1.23
Indicated[a]	1.76	1.74	1.95	1.95	1.92	1.95	1.94
Theoretical	2.82	—	—	—	—	—	2.61
Compression Ratio	7.90	7.79	7.66	7.51	7.64	7.63	7.78
Suction Pressure (psia)	19.2	22.3	25.4	29.4	31.0	31.3	31.3
Discharge Pressure (psia)	151	174	195	221	237	239	243
Suction Temperature (F)[b]	125	123	122	121	126	125	128
Discharge Temperature (F)[b]	233	238	244	252	265	266	272
Motor Input (watt)	625	680	700	780	815	820	823
Evaporator Input (watt)	572	660	781	936	991	1,016	1,013

[a] $\text{Indicated C.O.P.} = \dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions} - \text{Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

Table II-C—Performance Characteristics of Refrigerants 22/12 Mixtures

Nominal Conditions: 40 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor A

	Data for Indicated Refrigerants 22/12 Mixtures (wt %)						
	0/100	12/88	25/75	50/50	75/25	85/15	100/0
Refrigerating Capacity							
Measured (Btu/hr)	8,660	9,940	11,060	12,555	13,310	13,545[c]	13,660[c]
% of Refrigerant 12	100	115	128	145	154	156	158
% of Refrigerant 22	63	73	81	92	97	99	100
Coefficient of Performance							
Actual	2.86	3.00	3.22	3.28	3.31	3.27	3.28
Indicated[a]	4.34	4.35	4.60	4.49	4.43	4.34	4.35
Theoretical	5.91	—	—	—	—	—	5.64
Compression Ratio	2.93	2.85	2.83	2.83	2.86	2.86	2.90
Suction Pressure (psia)	51.7	60.7	68.9	78.0	82.7	83.5	84.9
Discharge Pressure (psia)	151	173	195	221	237	239	243
Suction Temperature (F)[b]	86	84	84	84	88	84	86
Discharge Temperature (F)[b]	170	171	173	178	190	187	191
Motor Input (watt)	885	970	1,005	1,120	1,180	1,215	1,220
Evaporator Input (watt)	2,537	2,912	3,240	3,678	3,900	3,969	4,003

[a] Indicated C.O.P. = $\frac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

[c] Voltage increased from 230 to 235 volt. This was necessary to obtain sufficient heater capacity at the evaporator. Compressor motor also operated at the higher voltage.

sure readings are taken directly from the test unit at P_2 and P_3. The compression ratio, of course, is the ratio of the latter to the former.

Both the suction and discharge gas temperatures reported are the averages of the readings measured on both cylinders with special thermocouples at T_5 and T_6, as described in Section A under Calorimeter Test Unit. In each case the temperatures measured were within 1 F of each other. They represent the best values known to the authors for such temperatures. It is estimated that they will be even better if 5-10 F are subtracted from the suction temperatures and a similar amount added to the discharge values. These corrections represent heat gains from, or losses to, the compressor housing due to radiation.

C. Accuracy of Results—The overall accuracy of the refrigerating capacity data obtained in this unit is dependent on the accuracy of the individual instruments. For instance, the motor-compressor is provided with a constant voltage supply of 230 volt ± 0.1%. The absolute pressure gage used to measure suction pressures up to 58 psia can be read accurately to 0.05 psi. Frequent calibration runs are made on the various thermocouples used to assure accuracy within ± 2

Table III-A—Performance Characteristics of Refrigerants 13B1/12 Mixtures

Nominal Conditions: —20 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor B

	Data for Indicated Refrigerants 13B1/12 Mixtures (wt %)				
	0/100	25/75	50/50	75/25	100/0
Refrigerating Capacity					
Measured (Btu/hr)	1,150	1,295	1,480	2,670	4,160
% of Refrigerant 12	100	112	129	232	362
% of Refrigerant 22	53	60	68	123	191
Coefficient of Performance					
Actual	0.58	0.62	0.67	0.94	1.10
Indicated[a]	1.20	1.22	1.24	1.48	1.51
Theoretical	2.50	—	—	—	2.02
Compression Ratio	9.88	9.89	9.68	8.50	7.30
Suction Pressure (psia)	15.3	17.0	19.8	31.5	48.6
Discharge Pressure (psia)	151	168	192	268	355
Suction Temeprature (F)[b]	131	129	125	117	106
Discharge Temperature (F)[b]	233	235	239	247	249
Motor Input (watt)	580	610	650	830	1,105
Evaporator Input (watt)	337	379	434	783	1,219

[a] Indicated C.O.P. = $\frac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

F in the temperature measurements made. Likewise, the electrical measurement devices have been checked to maintain their original accuracy of ±1%.

As a result of making numerous runs on Refrigerants 12 and 22, it has been found that replicate runs will be within 0.3% of each other. Each time a comparison run is made between refrigerants, a new calibration run is made on the base refrigerant (usually 12 or 22).

Considering the entire system as a unit, it is estimated that the refrigerating capacity data obtained have an over-all accuracy of ±5% or better. In making this estimate consideration was given to the effect on refrigerating capacity of a small subcooling of the liquid that occurred in the line between the receiver (D) and the expansion valve (E). It was found to represent an increase in capacity of from 2% at most liquid flow rates to about 4% at a few of the very lowest flows. None of the cooling capacity values has been corrected for this effect.

D. Refrigerants Studied—The individual compounds or azeotropes tested in this work were Refrigerants 12, 13B1, 22, 115, 500 (the azeotrope 12/152a, 73.8/26.2 wt %), and 502 (the azeotrope 22/115, 48.8/51.2 wt %). The results ob-

Table III-B—Performance Characteristics of Refrigerants 13B1/12 Mixtures

Nominal Conditions: —10 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor B

	Data for Indicated Refrigerants 13B1/12 Mixtures (wt %)				
	0/100	10/90	25/75	65/35	100/0
Refrigerating Capacity					
Measured (Btu/hr)	1,950	2,045	2,555	3,855	5,540
% of Refrigerant 12	100	105	131	198	284
% of Refrigerant 22	56	59	74	112	160
Coefficient of Performance					
Actual	0.92	0.89	1.00	1.21	1.31
Indicated[a]	1.76	1.60	1.66	1.79	1.74
Theoretical	2.02	—	—	—	2.35
Compression Ratio	7.90	7.88	7.33	6.89	5.94
Suction Pressure (psia)	19.2	21.3	26.2	38.9	59.7
Discharge Pressure (psia)	151	168	192	268	355
Suction Temperature (F)[b]	125	118	113	105	100
Discharge Temperature (F)[b]	233	229	232	235	238
Motor Input (watt)	625	675	750	930	1,235
Evaporator Input (watt)	572	599	749	1,129	1,623

[a] Indicated C.O.P. = $\dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

tained are given in Tables I-A,B,C along with similar data for single mixtures of Refrigerants 13/12 and 13/22.

Numerous mixtures of Refrigerants 12 and 22 with each other and with Refrigerant 13B1 were examined. The results are given in Tables II-A,B,C, III-A,B,C and IV-A,B,C. The refrigerating capacity data for the three mixtures at the −10 F nominal evaporating temperature also are plotted in Fig. 2. Similar curves can be prepared for the −20 F evaporator conditions by referring to the data of Tables II, III and IV.

Table V contains representative cooling capacity data at three evaporator temperatures over a range of capacities from that of Refrigerant 12 to that of Refrigerant 13B1. The values listed are taken from Tables I-IV. The capacity data also are reported here in terms of their percentage of the capacities of Refrigerants 12 and 22.

A comparison has been made of the effect of evaporator temperature on the refrigerating capacity of Refrigerants 12 and 22. This is shown in Fig. 3 where curves obtained from both measurements and theoretical calculations are given.

Table III-C—Performance Characteristics of Refrigerants 13B1/12 Mixtures

Nominal Conditions: 40 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor B

	Data for Indicated Refrigerants 13B1/12 Mixtures (wt %)				
	0/100	10/90	25/75	65/35	100/0
Refrigerating Capacity					
Measured (Btu/hr)	8,660	9,080	10,230	6,595[c]	8,125[c]
% of Refrigerant 12	100	105	118	—	—
% of Refrigerant 22	63	67	75	110	135
Coefficient of Performance					
Actual	2.86	2.86	3.00	2.33	2.29
Indicated[a]	4.34	4.22	4.28	3.65	3.22
Theoretical	5.91	—	—	—	5.14
Compression Ratio	2.93	2.91	2.87	2.71	2.56
Suction Pressure (psia)	51.7	57.7	66.7	98.7	139
Discharge Pressure (psia)	151	168	192	268	355
Suction Temperature (F)[b]	86	84	82	100	95
Discharge Temperature (F)[b]	170	169	168	178	180
Motor Input (watt)	885	930	1,000	830	1,040
Evaporator Input (watt)	2,537	2,660	2,997	1,932	2,380

[a] Indicated C.O.P. = $\dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions} - \text{Motor watt at no-load conditions (300)}}$.

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

[c] Compressor operating on but 1 cyl.

A few runs were made with Refrigerants 13B1 and 22 at nominal evaporator temperatures of −60, −40 and 0 F to show their relative performance characteristics, especially at the lower temperatures. The results are reported in Table VI.

E. Use of Data – A wealth of information is presented in the tables. Their single most important value is in comparing the cooling capacities of the various refrigerants studied. For example, in the case of any given compressor the data available make it possible to pick alternate refrigerants for use in it which will give any desired reasonable increase in its capacity. By selecting the proper compounds or mixtures the same capacity can be obtained in a unit operating on either 50 or 60-cycle current. By choosing the proper series of refrigerants it is possible to obtain a maximum of flexibility in cooling capacity with a minimum number of compressor sizes. More than one size of motor will be required for each compressor since any increase in capacity requires an increase in motor power input.

In using the data presented it should be remembered that all the values were obtained with a single

Table IV-A—Performance Characteristics of Refrigerants 13B1/22 Mixtures

Nominal Conditions: —20 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor B

	Data for Indicated Refrigerants 13B1/22 Mixtures (wt %)				
	0/100	25/75	50/50	75/25	100/0
Refrigerating Capacity					
Measured (Btu/hr)	2,175	2,850	3,250	3,725	4,160
% of Refrigerant 12	189	248	283	324	362
% of Refrigerant 22	100	131	149	171	191
Coefficient of Performance					
Actual	0.84	1.03	1.00	1.04	1.10
Indicated[a]	1.38	1.63	1.46	1.45	1.51
Theoretical	2.30	—	—	—	2.02
Compression Ratio	9.74	9.13	8.44	7.80	7.30
Suction Pressure (psia)	25.0	30.2	36.6	43.6	48.6
Discharge Pressure (psia)	243	276	309	340	355
Suction Temperature (F)[b]	141	134	125	115	106
Discharge Temperature (F)[b]	282	279	273	261	249
Motor Input (watt)	760	813	950	1,050	1,105
Evaporator Input (watt)	637	835	952	1,091	1,219

[a] Indicated C.O.P. = $\frac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

model of compressor. The same relationships reported here will apply to any other compressor even though the actual capacity values may be shifted up or down. To obtain absolute values it is necessary to carry out measurements with the specific compressor under consideration.

Some typical examples of the increased capacity that can be obtained by using a lower-boiling refrigerant or by adding a lower-boiling component to refrigerants such as 12 or 22 are shown in Table V. The refrigerants are listed in the order of their increasing capacities at the −20 F evaporator conditions. From these data the engineer who has the problem of obtaining increased capacity from a given compressor can determine what possible compounds and mixtures will apply to his case. Thus, an examination of Table V will show him that in obtaining a 30% increase in capacity over that of Refrigerant 12 he has a choice of any one of three mixtures of refrigerants; namely, 13/12 (11/89 wt %), 13B1/12 (25/75 wt %), or 22/12 (25/75 wt %).

Table V also shows comparative data at evaporator temperatures of −10 and 40 F. It will be noted that in some cases the rates of increase in refrigerating capacity are quite different from those at

Table IV-B—Performance Characteristics of Refrigerants 13B1/22 Mixtures

Nominal Conditions: —10 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Tests made with Compressor B

	Data for Indicated Refrigerants 13B1/22 Mixtures (wt %)				
	0/100	25/75	50/50	75/25	100/0
Refrigerating Capacity					
Measured (Btu/hr)	3,455	4,145	4,555	5,145	5,540
% of Refrigerant 12	177	213	234	264	284
% of Refrigerant 22	100	120	132	149	160
Coefficient of Performance					
Actual	1.23	1.25	1.25	1.29	1.31
Indicated[a]	1.94	1.81	1.75	1.74	1.74
Theoretical	2.61	—	—	—	2.35
Compression Ratio	7.78	7.29	6.86	6.39	5.94
Suction Pressure (psia)	31.3	37.8	45.0	53.1	59.7
Discharge Pressure (psia)	243	276	309	340	355
Suction Temperature (F)[b]	128	121	112	105	100
Discharge Temperature (F)[b]	272	267	258	248	238
Motor Input (watt)	823	970	1,065	1,165	1,235
Evaporator Input (watt)	1,013	1,215	1,335	1,507	1,623

[a] Indicated C.O.P. = $\dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions} - \text{Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

–20 F. In fact, the order of increase is even reversed. For example, the capacity of Refrigerant 115 is 16% greater than that of the 25/75 wt % mixture of Refrigerants 22/12 when each is compared to 12 at the –20 F evaporator temperature. At –10 F evaporator, on the other hand, Refrigerant 115 has 9% less capacity than does the Refrigerant 22/12 mixture. This is a startling difference and may have real significance to some specific application.

With any mixture the refrigerating capacity can be increased from that of the high-boiling component to that of the low-boiling one. For example, at an evaporator temperature of –10 F and a condenser temperature of 110 F, Refrigerant 12 has a cooling capacity of 1950 Btu/hr in our specific calorimeter unit while Refrigerant 13B1 has a capacity of 5540 Btu/hr (Table III-B). Any desired capacity between these two values can be obtained by using the proper mixture of the two compounds. The curves of Fig. 2 are helpful in making such a choice not only for Refrigerants 13B1/12 mixtures but also for those of Refrigerants 22/12 and 13B1/22.

It is well known that refrigerating capacity is increased as the

Table IV-C—Performance Characteristics of Refrigerants 13B1/22 Mixtures

Nominal Conditions: 40 F Evaporating temperature
65 F Temperature of vapor entering compressor
110 F Condensing temperature

Compressor operating on only one cylinder

Tests made with Compressor B

	Data for Indicated Refrigerants 13B1/22 Mixtures (wt %)				
	0/100	25/75	50/50	75/25	100/0
Refrigerating Capacity					
Measured (Btu/hr)	6,010	6,910	7,800	8,170	8,125
% of Refrigerant 22	100	115	130	136	135
Coefficient of Performance					
Actual	2.20	2.30	2.44	2.37	2.29
Indicated[a]	3.52	3.49	3.60	3.37	3.22
Theoretical	5.64	—	—	—	5.14
Compression Ratio	2.91	2.79	2.71	2.62	2.56
Suction Pressure (psia)	83.7	98.7	114	130	139
Discharge Pressure (psia)	243	276	309	340	355
Suction Temperature (F)[b]	110	105	97	95	95
Discharge Temperature (F)[b]	208	201	190	184	180
Motor Input (watt)	800	880	935	1,010	1,040
Evaporator Input (watt)	1,761	2,025	2,285	2,394	2,380

[a] Indicated C.O.P. = $\dfrac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

Table V—Comparison of Refrigerating Capacities

Nominal Operating Conditions: 110 F Condenser
65 F Vapor entering compressor

Refrigerant (Compositions in wt %)	Boiling Point (F)	−20 F Evaporator: Compressor Used	−20 F Evaporator: Refrig. Cap. (Btu/hr)	−20 F Evaporator: % of Cap. of 12	−20 F Evaporator: % of Cap. of 22	−10 F Evaporator[a]: Refrig. Cap. (Btu/hr)	−10 F Evaporator[a]: % of Cap. of 12	−10 F Evaporator[a]: % of Cap. of 22	40 F Evaporator[a]: Refrig. Cap. (Btu/hr)	40 F Evaporator[a]: % of Cap. of 12	40 F Evaporator[a]: % of Cap. of 22
12	−21.6	A	1,250	100	53	1,950	100	56	8,660	100	63
		B	1,150	100	53						
500	−28.0	A	1,525	122	65	2,375	122	69	10,155	117	74
13/12 (11/89)		B	1,470	128	68	2,340	120	68	10,620	123	78
13B1/12 (25/75)		B	1,480	129	68	2,555	131	74	10,230	118	75
22/12 (25/75)		A	1,625	130	69	2,665	137	77	11,060	128	81
115	−37.7	B	1,675	146	77	2,505	128	73	10,040	116	73
22/12 (50/50)		A	2,135	171	90	3,195	164	93	12,555	145	92
22	−41.4	A	2,360	189	100	3,455	177	100	13,660	158	100
		B	2,175	189	100						
502	−50.1	A	2,650	212	112	3,780	194	110	14,090	163	103
13/22 (10/90)		B	2,500	217	115	3,755	193	109	7,400[b]	—	123
13B1/12 (65/35)		B	2,670	232	123	3,855	198	112	6,595[b]	—	110
13B1/22 (25/75)		B	2,850	248	131	4,145	213	120	6,910[b]	—	115
13B1/22 (50/50)		B	3,250	283	149	4,555	234	132	7,800[b]	—	130
13B1	−72.0	B	4,160	362	191	5,540	284	160	8,125[b]	—	135

[a] Equivalent results are obtained with either Compressor A or B
[b] Compressor operating on only 1 cyl

Table VI—Performance Characteristics of Refrigerants 13B1 and 22

Tests made with Compressor A

Nominal Conditions								
Evaporator (F)	—60		—40	—40		—40		0
Vapor Entering Comp. (F)	65		65	—20		65		65
Condenser (F)	90		110	90		90		85
Refrigerant	13B1	22	13B1	13B1	22	13B1	22	13B1
Refrigerating Capacity								
Measured (Btu/hr)	825	[c]	1,930	1,950	750	2,790	5,865	9,580
% of Refrigerant 22	—	—	—	—	100	372	100	163
Coefficient of Performance								
Actual	0.38	—	0.69	0.65	0.41	0.99	2.06	2.33
Indicated[a]	0.72	—	1.09	0.99	0.94	1.56	3.21	3.10
Theoretical	1.68	1.77	1.68	2.55	2.21	2.26	4.37	4.15
Compression Ratio	14.2	15.9	11.1	8.62	12.1	8.61	4.43	3.62
Suction Pressure (psia)	19.4	15.3	31.9	31.9	15.3	31.9	38.8	71.2
Discharge Pressure (psia)	275	243	355	275	185	273	172	258
Suction Temperature (F)[b]	147	—	130	47	156	120	105	86
Discharge Temperature (F)[b]	259	—	274	190	268	254	225	192
Motor Input (watt)	635	—	820	875	535	823	835	1,205
Evaporator Input (watt)	241	—	566	571	220	817	1,718	2,807

[a] Indicated C.O.P. = $\frac{\text{Evaporator watt input}}{\text{Motor watt at load conditions — Motor watt at no-load conditions (300)}}$

[b] Refrigerant temperatures in compressor head approximately 1/16 in. from valve reeds.

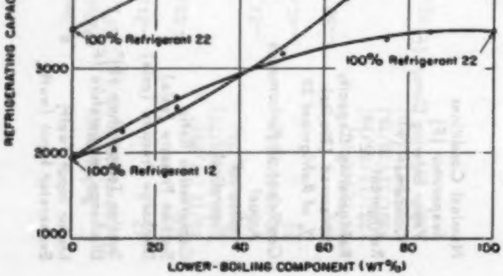
[c] Refrigerating capacity too low for measurement in test unit (less than 400 Btu/hr).

evaporator temperature is raised while the other operating conditions are kept constant. However, there are large differences between the theoretical values and those actually obtained. These differences are illustrated by the curves for Refrigerants 12 and 22 shown in Fig. 3. It is seen that actual refrigerating capacities are always less than the theoretical values and that, at low evaporator temperatures, the relative percentage loss of capacity is much greater than at high evaporator temperatures. This is due primarily to the lower volumetric efficiency of the compressor at the high compression ratios associated with the lower temperatures.

The performances of Refrigerants 13B1 and 22 are compared under a few lower-temperature conditions (Table VI). The data illustrate the greater capacity of the lower-boiling Refrigerant 13B1. For example, it has over 3 times the cooling capacity of Refrigerant 22 at evaporator and condenser temperatures of –40 and 90 F, respectively. This is due to the greater suction pressure and lower compression ratio of Refrigerant 13B1.

Another property of refrigerants that is of concern to the engineer is the temperature of the discharge gas at a given compression ratio. It is interesting to note the differences in the values obtained for the various compounds and azeotropes studied. From Tables I-A,B,C we see that, at any given set of conditions, Refrigerant 22 has the highest discharge gas temperature and Refrigerant 115 the

Fig. 2 Effect of refrigerant composition on refrigerating capacity

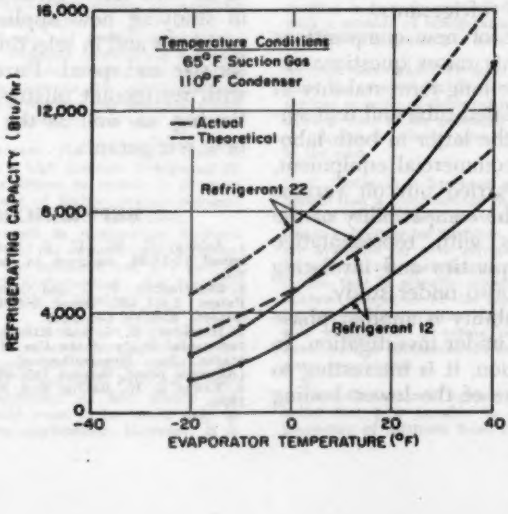


lowest of the compounds studied. In the case of Refrigerant 502, an azeotrope of Refrigerants 22 and 115, the 115 serves to reduce the discharge gas temperature. At the same time there is a small increase in refrigerating capacity over that of Refrigerant 22. This may be of real importance to some applications.

F. Azeotropes and Mixtures — An azeotrope is a mixture, usually of two compounds, which behaves physically as if it were a single pure compound. Thus, its vapor composition is the same as that of the liquid. This is true, however, for only a specific composition at a specific pressure. Azeotropes with boiling points below those of either component, as illustrated by the two azeotropes discussed in this work, have refrigerating capacities greater than those of the individual components. For example, the cooling capacity of Refrigerant 502 is greater than that of either Refrigerants 22 or 115 (Tables I-A,B,C). In contrast, an ordinary mixture (which is not an azeotrope) has a capacity between those of its components.

In an actual refrigerating unit other factors besides capacity must be considered. During operation, we have shown that both azeotropes and mixtures have fixed reproducible cooling capacities. In both cases, the composition of the gas being circulated is the same as that of the refrigerant charged. On shutdown, however, the low-boiling component of a mixture will concentrate in the vapor phase due to its higher vapor pressure.

Fig. 3 Effect of evaporator temperature on refrigerating capacity



An azeotrope is subject to a similar effect though to a lesser degree, because its composition will change with change in the pressure of the system. The exact amount of this effect will depend on the characteristics of the specific azeotrope.

The above comments apply only to systems using flash-type evaporators. The use of mixtures in flooded evaporators has not been studied.

Another factor is the relative oil solubility of the components of the azeotrope or mixture. Generally, the lower-boiling component is more highly fluorinated and thus has poorer solubility in the oil. This accentuates its concentration in the vapor phase, regardless of whether it is present as part of an azeotrope or of a mixture.

G. Continuing Program – Our calorimeter test program is continuing in the search for compositions having singular properties. It also being extended some low-temperature refrigerants.

The use of new compositions as refrigerants raises questions regarding their long-term stability in a system. Sealed-tube and unit stability tests, the latter in both laboratory and commercial equipment, are being carried out on various mixtures. The compatibility of the compositions with representative elastomers, plastics and insulating materials is also under study.

Oil solubility is another phase of the work under investigation. In this connection, it is interesting to note that one of the lower-boiling compounds studied, Refrigerant 13B1, has good solubility in oil. For example, its solubility is far better than that of Refrigerant 22.

SUMMARY

The performance of a variety of refrigerants, azeotropes and mixtures has been studied in a calorimeter unit and is reported here. Comparative capacity data have been obtained over a range of temperatures typical of refrigerating and air-conditioning applications. These studies show that the use of mixtures is both feasible and practical and that definite and reproducible operation can be obtained with them. Actual operating information provides a basis for further consideration by refrigerating engineers.

This practical demonstration of the use of mixed refrigerants provides almost unlimited possibilities for varying the capacity of refrigerating systems. Design engineers will have greater flexibility in studying new applications and problems and in selecting compressor size and speed. Further studies with refrigerant mixtures are continuing, as well as the search for new refrigerants.

REFERENCES

1. Ashley, C. M., U. S. Patent 2,492,725, issued 12-27-49, assigned to Carrier Corporation.
2. Etherington, T. L. and Cannon, P., U. S. Patent 2,951,350, issued 9-6-60, assigned to General Electric Company.
3. Haselden, G. G. and Klimek, L., "An Experimental Study of the Use of Mixed Refrigerants for Non-isothermal Refrigeration" (Advance proof, Session 1957-58).
4. Krans, D. H., Refrig. Eng. No. 12, 43 (Dec. 1954).

DISCUSSION

T. Atwood, Edgewater, N. J. (Written): The subject of mixed refrigerants has been frequently discussed but the doubts concerning changes in composition in the system and unequal loss of components via leaks have deterred any real study of such systems. This work should revive interest and, provided no substantial deleterious effects are discovered, may lead to tailoring refrigerants for systems instead of tailoring systems for available single component refrigerants.

Development of thermodynamic tables for mixtures is an almost impossible task and still would not give the practical answer. The method described is simple and does give a practical answer to the thermodynamic relationships in an operating system.

We also performed a series of experiments with Refrigerants 12 and 22 using ASHRAE Standard 23 R with 86/5 F cycle and approximately 20 deg superheat. Our tests indicated a somewhat greater capacity increase of mixtures passing through a maximum in the 40-60% Refrigerant 12 range with an accompanying increase in coefficient of performance. This may be due to variations in compressor efficiencies, and we would like to know the refrigerant for which the compressors were designed in the tests described by Mr. Chapman. Our experimental set-up was similar using a dry expansion cycle. We wish to confirm that refrigerant composition does not change in the system; samples were taken before and after the compressor and before and after the expansion valve.

The differences between two identical compressors on some cycles are quite startling and certainly point up the difficulty in achieving reproducibility of results in different compressor designs. It would be of interest to have the same tests run with a series of compressors of comparable displacement, but designed for different refrigerants.

The discharge and suction temperatures obtained from locations immediately over the valve plates are of real interest. The difference in temperature between the valve area and the conventional thermometer locations would be of general interest if they were available.

Author Chapman: The compressor used in these tests was a high pressure Refrigerant 22 type. This was chosen to enable us to carry out investigations of higher-pressure refrigerants in the range of a 40 F evaporator.

The difference in temperature between the valve area and the conventional thermometer locations was measured and is a part of the test record. Unfortunately, this information is not available at present.

H. M. Elsey, Oakmont, Pa. (Written): It is my belief that the most interesting of the compounds, azeotropes and ordinary mixtures reported upon is Refrigerant 13B1 or monobromotrifluoromethane. At first glance this refrigerant would seem to be useful only in low-temperature applications. However, it is stated to have a greater solvent power for oils than Refrigerant 22. Further tests should show that the reactivity of Refrigerant 13B1 is satisfactorily low and that it will have a lesser softening action on wire enamels and motor impregnating varnishes than does Refrigerant 22. It may even approach Refrigerant 12 with respect to this property. It is indicated that it will have an advantage over Refrigerant 22 in returning oil to the compressor. These advantages over Refrigerant 22, together with the use of a smaller compressor which would go with a denser gas, might be sufficient to warrant the design of compressors specifically for the use of Refrigerant 13B1 in air-conditioning systems. However, there would be greater difficulty in keeping the noise down to a satisfactory level.

As a chemist, I am not enthusiastic about using mixtures of Refrigerants 12 and 22 in refrigeration systems. Each refrigerant has a different set of problems which must be designed around.

Refrigerant mixtures as a class do not appeal to me. A mixture is used to increase or decrease the capacity of an existing compressor by a desired percentage under given operating conditions. The desired capacity will only be obtained if the refrigerant charge remains constant. Refrigerant leaks are not unknown. Since it is more concentrated in the gas phase, the more volatile constituent will escape more rapidly from a leaky system and the capacity will change continuously until the leak is repaired and the machine recharged. Also, frequently the air is purged out of a system along with some of the refrigerant rather than by pumping it out with a vacuum pump. Removal of air by purging with the refrigerant can be a satisfactory process when the refrigerant is a single compound and the purging is properly controlled. Purging of air with a mixture of refrigerants can change the capacity of a system just as a major leak will.

Author Chapman: We do not advocate complete use of mixtures. We are suggesting that the use of some of these mixtures may help in solving some of the problems of the compressor designers which have not been solved in the past.

C. T. Thomas: It has been stated by several people that a disadvantage in the use of mixtures is the fact that there are no thermodynamic properties for mixtures, while good data are available for pure refrigerants. It should be pointed out that the normal refrigeration system that is actually working with a fluid mixture comprised of the lubricating oil and the refrigerant can cause changes of considerable magnitude. It seems possible that we could predict properties for mixtures from the pure components.

Author Chapman: I agree with Mr. Thomas' belief that it is possible to estimate certain properties of mixtures from the properties of

the pure components. We have used this procedure regularly to scout, for instance, the vapor pressure of a mixture at a given temperature or the theoretical refrigerating capacity of a mixture prior to running the mixture in our calorimeter.

W. Holladay, Los Angeles, Calif.: Refrigerant 13B1 has an evaporating range temperature between minus 40 and minus 65 where, in the last few years, it has been necessary to use either cascade or Compound F12 or Refrigerant 12 or 22. Compounding and cascading is expensive; it is also troublesome from an operating standpoint. Refrigerant 13B1, with an oversized condenser, does a very satisfactory job.

Walter O. Walker, Coral Gables, Fla.: Was any effort made to use a fluid type evaporator? Were any efforts made to make runs on using a fluid evaporator? Were there any speculations as to how fluid evaporator temperatures would relate themselves, or rather, the refrigerant would relate itself in composition of the circulating gas?

Author Chapman: No, we have not done this.

W. O. Walker: I want to commend the authors for including the tube test of the mixtures of azeotropes with the oils and other materials. Without these tests to demonstrate the stability of the refrigerant oil mixtures in the presence of materials in the refrigerating system, little practical value can come from a determination of the refrigerating capacity of the characteristic mixtures and azeotropes. If there is no stability, the azeotropes cannot be used under other than specified conditions.

1759

No. 1759

Solar Heat Gains through Domed Skylights

L. F. SCHUTRUM
Member ASHRAE

N. OZISIK

Solar heat gains through four types of light controlling and three types of flat glass skylights were reported upon some time ago by Messrs. Vild and Parmelee[1]; however, information on the heat gains through a quite common type skylight, the plastics dome, which is becoming increasingly popular, was not available. The purpose of this study was to obtain data so that solar heat gains through acrylic plastics dome skylights could be determined by the air-conditioning engineer. Three domes having high, medium and low light transmitting qualities were selected to cover the range of normal practice, and a fourth dome which was painted black was tested in the ASHRAE Solar Calorimeter.[2]

This calorimeter has been used over the years for compiling data on heat flow through fenestration under the guidance of the Society's Technical Advisory Committees. The present study was started under the guidance of the former TAC on Heat Flow Through Fenestration, which has been discontinued through reorganization of the Technical Committees.

TEST APPARATUS

The heat gain through the skylights was determined by means of the solar calorimeter, from the rate of flow of the circulating mixture of ethylene glycol and water, and the difference between inlet and outlet fluid temperatures. Adjustments were made for the heat flow through the back of the calorimeter. The calorimeter could be rotated to any desired azimuth and could be tilted through 90 deg about a horizontal axis. The opening to which the skylight was fitted was 44¼ x 44¼ in.

Instrumentation – The solar radiation was measured with two Eppley pyrheliometers mounted on the

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face of the panel on opposite sides of the calorimeter opening. A third pyrheliometer was used at times to measure the direct normal solar radiation. Other weather conditions recorded were of the dry bulb and wet bulb temperatures, wind velocity and direction, and an indication of the cloudyness and hazyness of the sky. A convection compensated radiometer[3] was used to measure the low temperature radiation coming from the sky. The temperatures of the calorimeter, skylight and outdoor air were sensed by thermocouples, and as much as possible of the data was recorded on a 16-point electronic roll chart recorder. Manual readings also were taken by means of an electronic potentiometer.

Skylights—The four plastics domes and one ceiling light diffuser which were used in these tests were made from 4 by 4 ft sheets of acrylic plastics to fit the opening of the calorimeter (44¼ x 44¼ in.). Three domes, and the diffuser being held by the technician, are shown in Fig. 1. The dome height at the center was approximately ¼ of the span, and the height of the light diffuser was about 2 in. Two of the domes shown in the picture are light-diffusing types, and the third was painted black for special test purposes. A fourth dome, which is transparent, is shown in Fig. 2, in position on top of the calorimeter ready for testing. In this case it is mounted on a special curbing, 18 in. high, but in other tests a 9-in. curb or no curbing at all was used. Also seen in Fig. 2 are two pyrheliometers and the convection compensated radiometer. A brief description of the domes is given in Table I.

Test Procedure — The solar calorimeter was set in a horizontal position for most of the tests except where special angles of incidence were desired, in which case the calorimeter was tilted. Tests were made with the calorimeter in a fixed position or with the calorimeter adjusted to follow the sun. Data were gathered throughout the 10 or 15-min test periods, on the solar radiation, desired angles, outdoor environment, calorimeter liquid flow rate and temperature

Fig. 1 Plastics domes and light diffuser

rise, and the various surface temperatures.

Analysis—Solar energy falling upon a dome skylight will be reflected partly from the dome, partly absorbed, and the remaining radiation will pass on through into the interior of the building. The reflected radiation is rejected and is not involved in the heat balance. Because the skylight is heated by the absorbed radiation, it will increase in temperature until the rate of solar radiation absorbed equals the rate of heat loss by radiation and convection to the outdoors and indoors, and by conduction heat flow to the supporting members. The radiation and convection and the aforementioned solar radiation passing through the skylight are the components which constitute the total solar heat gain.

METHOD OF ATTACK

At the outset, it was decided to compute the solar properties of the domes to be tested from measured properties of the flat plastics sheets from which the domes were formed. These theoretical properties of the domes could then be verified by experimentation in the solar calorimeter and re-evaluated if necessary. Convection and radiation exchanges between the skylight and interior would be determined by test.

Transmittance—The solar transmittance is defined as the ratio of the rate of solar energy passing through a material and the incident solar energy. Reflectance similarly is a ratio of reflected to incident radiation. The solar transmittances and reflectances of flat sheets of three thicknesses and of the three basic types of plastics were measured by means of an Eppley pyrheliometer. These measurements were made for angles of incidence up to 60 deg. The trans-

Fig. 2 Transparent dome and 18-in. curbing mounted on solar calorimeter for testing

mittances and reflectances of flat sheets for various angles of incident solar radiation are given in Appendix A.

From the transmittances and reflectances of the flat sheets, the solar transmittances and reflectances were computed for a hemispherical shaped dome and a spherical segment (Fig. 3). The method of computation is described in the appendix, but essentially these properties were computed by dividing the dome into zones, having a limited range of incidence angles, and using the transmittances and reflectances determined from the flat sheets for the average incident angle of each zone. For the transparent dome, the transmitted radiation and radiation reflected from the outside of the dome were assumed to be specular. After the energy had entered through the dome any reflections from the dome surfaces were considered diffuse. For the diffusing domes both transmittances and reflectances were considered diffuse in nature.

Dome Skylights (No Curbing)—The computed transmittances of three types of flat plastics sheets, hemispherical domes and spherical segments are shown by the curves of Fig. 3. Note first of all that the transmittance is based on the normal intensity of the direct solar radiation instead of the incident radiation. This relationship was used so that the transmittance at a 90-deg incidence angle could be finite instead of being infinite (resulting from the incident radiation on a horizontal plane being zero). Experimental points determined from calorimeter tests of actual domes made of three types of plastics also are plotted in Fig. 3.

The calorimeter measured the total heat transfer which occurred through the domes. By subtracting the convection-radiation heat exchange between the dome and the calorimeter, and the diffuse radia-

Table I—Specific Features of Domes

Dome Designation	Solar Transmittance of Flat Sheet (Direct Normal)	Plastics[a]	Thickness of Base Sheet in.	Dome Height in.[b]
I Clear (Transparent)	0.86	Acrylic Colorless	3/16	11¼
II Medium Transmittance (Translucent)	0.52	Acrylic White W-2447	3/16	11¼
III Low Transmittance (Translucent)	0.27	Acrylic White W-7328	3/16	11¼
Light Diffuser (Translucent)	0.58	Acrylic White	1/8	2⅓

[b] All domes 44¼ x 44¼ in. at the base
[a] Number designation by Rohm & Haas Company

tion component, the transmittances for direct solar radiation were obtained. These convection-radiation and diffuse radiation corrections are a small part of the total gain for near-normal incidence angles, but become a major part of the total gain at high incidence angles where the direct component is quite small. It will be noted in Fig. 3, that the experimental points correlate best with the curve for the segment (height to diam = 0.21) which corresponds closely with the shape of the actual domes (height to width = 0.25). Tests also were

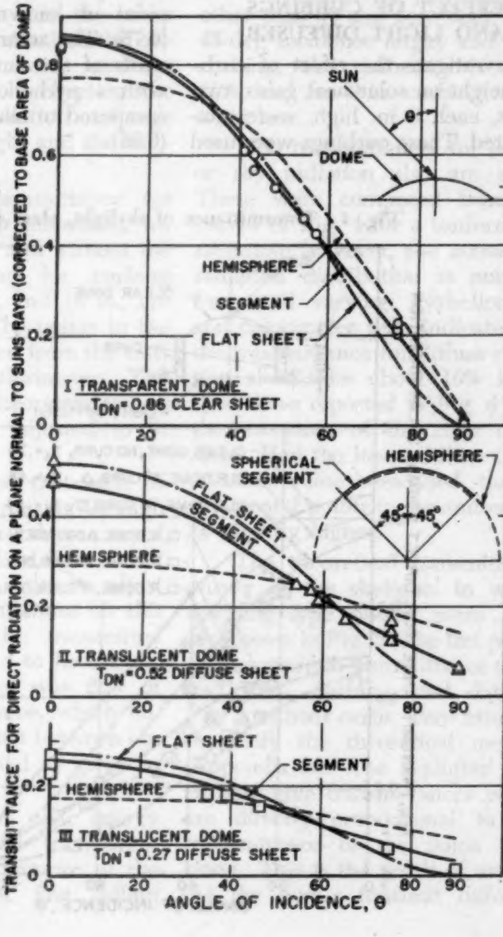


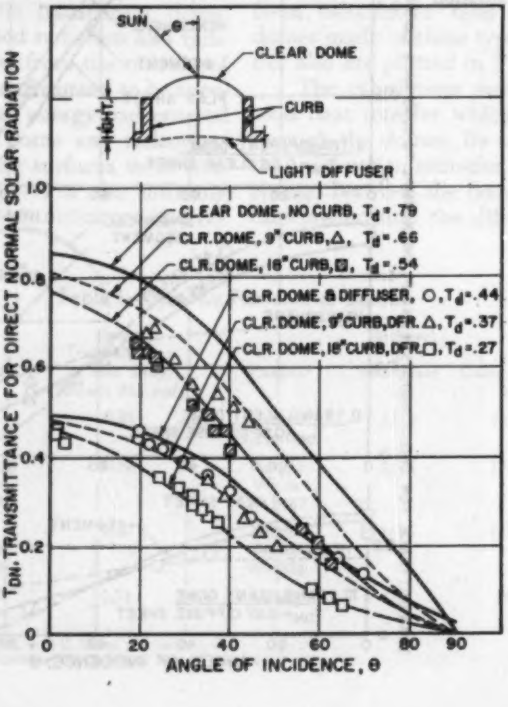
Fig. 3 Transmittance of domes for solar radiation

conducted with the calorimeter to determine the effect of the solar azimuth on the transmittances of the domes. The transmittances were found to be practically independent of the solar azimuth, i.e whether the sun's rays were in a plane parallel to the sides of the dome, or parallel to the diagonal.

EFFECT OF CURBINGS AND LIGHT DIFFUSER

To investigate the effect of curbing height on solar heat gains, two curbs, each 9 in. high, were constructed. These curbings were used to give curb heights of 9 in. and combined to give 18 in. The corresponding ratios of width to height are approximately 5 and 2½. The curbings were well insulated to reduce heat flow through the sides to a minimum and instrumented so that this heat leakage could be measured. The inside surfaces were painted with three coats of paint of known light reflectance (0.75). The solar reflectance of the walls of the curbing as measured with a pyrheliometer was 0.7 as compared to magnesium carbonate (0.95).

Fig. 4 Transmittance of skylight, clear dome



The solar transmittances of skylights of various combinations of dome, curb and the light-diffuser were computed from the known solar properties of the components (90-deg segment properties for the dome) by means of the transfer-function method given in reference 4 as described in the appendix. At the same time, tests were conducted with the solar calorimeter to substantiate the calculated transmittances, and to evaluate the factors effecting the convection and radiation exchange between the skylight and the calorimeter.

The solar transmittance for direct normal solar intensities, for a clear dome with and without the light diffuser, and for curbing heights of zero, 9, and 18 in., are given in Fig. 4. The points in the figure were obtained from the tests with the solar calorimeter. The curves shown are theoretical values which have been adjusted to fit the data better.

It was observed that some of the solar radiation passing through the edges of the clear dome caused high intensity bands of light to fall in the curbing. In order to adjust the theoretical calculations for this effect, some of the transmitted energy was assumed to be diffuse. The amount chosen was 25% of the total transmittance, which improved the correlation between observed and calculated as given in Fig. 4.

The amount of solar energy passing through the transparent dome and into the interior of the calorimeter without first falling upon the curb depends also, of course, upon the solar azimuth, because the curb in these tests was square in shape. The solar transmittances of the skylights given in the figure are for zero solar azimuth (sun in plane perpendicular to side). For the solar azimuths of 45 deg (diagonal), the transmittances are lower for angles of incidence other than 90 and zero deg. For a 45-deg incidence angle, and 18-in. curbing, the reduction is about 10%, and for 9-in. curbing, somewhat less.

The transmittances for diffuse or sky radiation also are given. These were computed from the curves of Fig. 4 for a uniform sky radiation, however, the actual sky radiation distribution is non-uniform and varying. Pyrheliometer and calorimeter tests indicate that the transmittance for diffuse radiation should be about 10% lower than those reported in Fig. 4 with the exception of the clear dome alone. Had the lower diffuse transmittance value been used, the experimental points in general would be slightly higher.

The theoretical transmittance curves of the skylights in which diffusing-type domes were used are shown in Fig. 5. The test points for the medium transmittance dome and 18-in. curbing, and data of Fig. 3 without curbs, were assumed to verify the theoretical method (Appendix B). The skylights with curbing give transmittances which are directly proportional to the transmittance of the dome used alone. This is the result of assuming the domes transmit diffusely

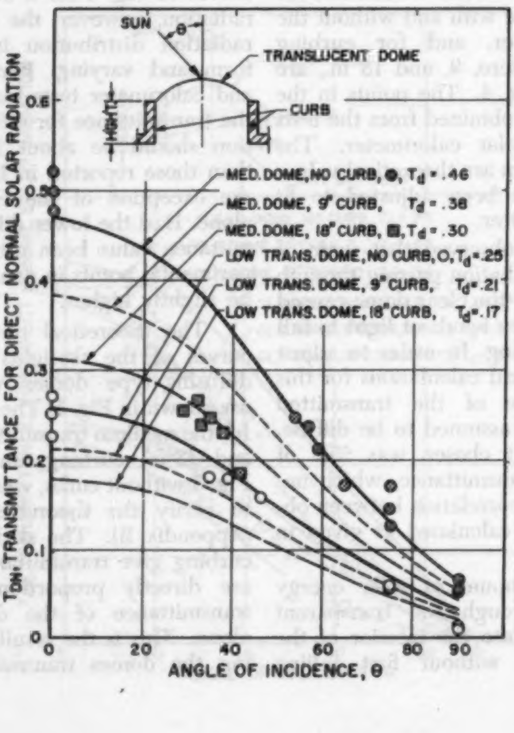
and that the curbing is also a diffuse reflector.

CONVECTION RADIATION GAIN

The convection and radiation heat exchange coefficients between the solar calorimeter and the skylight were determined by measurement without solar radiation entering the calorimeter. This was done by testing during the day with a black dome, which was opaque to solar radiation, or with the other transparent or translucent domes at night.

The convection heat exchange was determined by subtraction of the computed radiation exchange between skylight and calorimeter from the total calorimeter heat flow. A hemispherical emissivity of 0.83 was used for these calculations, which corresponds to a normal emissivity of about 0.875. Reference 6 gives the normal emissivity for plexiglas as 0.89. The convection heat transfer downward

Fig. 5 Transmittance of skylights, translucent domes



from the dome surface to the calorimeter was quite small, and under the test conditions for a horizontal skylight, the combined radiation-convection coefficient was about unity.

With the convection heat flow upwards from calorimeter-to-dome, the convection coefficient was of the order of 0.7 to 0.9 Btu/hr/sq ft/F, which gives a combined coefficient of about 1.4-1.9 Btu/hr/sq ft/F.

The heat exchange between the skylight dome and the outdoor, takes place by convection exchange with the outside air and low temperature radiation exchange with the sky and the surroundings which are seen by the dome. The radiant energy emission from cloudless skies can be estimated from the information given in reference 3 and was measured during many of the tests by a low temperature radiometer which is described in the same reference. A number of attempts were made to determine the convection heat exchange between the dome and outdoor air, by measuring the total loss from the dome at night by means of the calorimeter, and the radiant energy loss to the sky. The results were inconclusive.

Knowledge of the solar energy absorbed by the domes must be had in order that the portion of this energy which enters the room via convection and radiation can be determined. This absorptance was determined from the relationship of transmittance + reflectance + absorptance equals unity, since the first two solar properties were already determined and transmittance verified by experimentation.

The portion of the solar energy absorbed by a single-dome skylight (no curbings) which enters the space B is

$$C + R = U \frac{\alpha_{DM} I}{h_o} \tag{1}$$

and the heat entering due to a temperature difference between indoor and outdoor air is

$$C + R = U\ (t_o - t_i) \tag{2}$$

These equations can be combined if the appropriate U-value is used. The U-value does vary somewhat with the combined magnitudes obtained from equations 1 and 2.

$$C + R = U\left[\frac{\alpha_{DM} I}{h_o} + (t_o - t_i)\right] \tag{3}$$

In a similar manner the convection and radiation gain by the room from the skylight was related to the solar energy absorbed by the dome, curbing, and diffuser, where used, with the simplifying assumption that no heat flows through the curbing, and that thermal storage effects could be neglected.

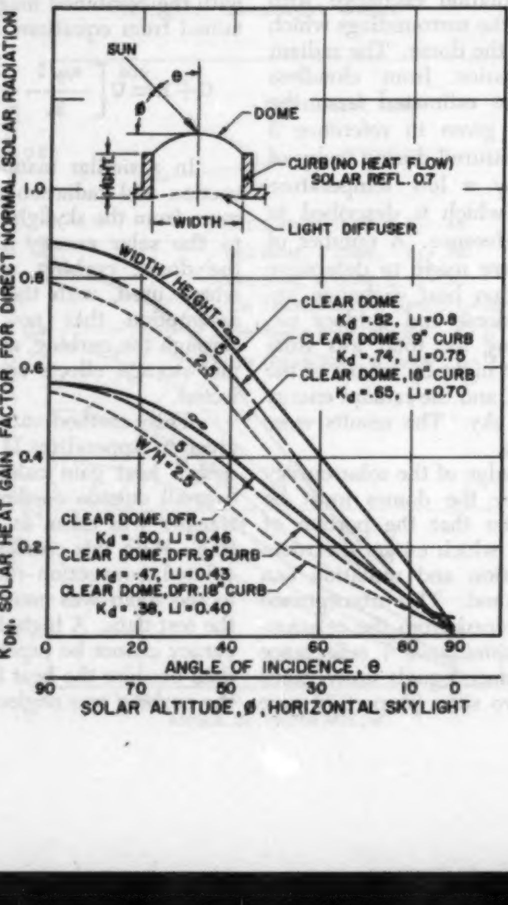
This method of solution is given in Appendixes D and E. For design heat gain calculations, the over-all outside coefficient of heat transfer was taken as 4 Btu/hr/sq ft/F. With this coefficient the calculated convection-radiation gain to the room was comparable with the test data. A high degree of accuracy cannot be expected in these data because the heat flow through the curbing was neglected in order

that the results would not be too complex for practical application.

DESIGN FACTORS

The solar energy transmitted through the skylight, and a portion of the convection-radiation gain to the room depend upon the solar radiation. To simplify the heat gain calculations solar heat gain factors K_D and K_d were developed in earlier papers.[9] For calculating the heat gains through plastic skylights, the K_D factor has been replaced by K_{DN} which must be multiplied by the normal intensity of the direct solar radiation rather than the incident direct radiation to determine the direct component of solar heat gain. The diffuse component may be determined by multiplying K_d by the incident diffuse radiation. The remaining part of the convection-radiation gain is

Fig. 6 Design factors for estimating solar heat gains through clear-dome horizonal skylights (See Eq. 3)



proportional to the indoor-outdoor temperature difference; consequently, the total gain can be given as:

$$\text{Total gain} = K_{DN} I_{DN} + K_d I_d + U (t_o - t_i) \quad (4)$$

These K factors and U-values suggested for estimating solar heat gains are given in Figs. 6 and 7.

Use of Design Factors – The K values are intended to give the air-conditioning engineer a means of estimating the solar heat gains through these plastics skylights for almost any location. Suppose one wishes to know the solar heat gain through a horizontal skylight having a clear dome and light diffuser with a ratio of width to height of about 5. The performance curve of this skylight is given in Fig. 6 (second curve from bottom). For

Fig. 7 Design factors for estimating solar heat gains through translucent-dome horizontal skylights

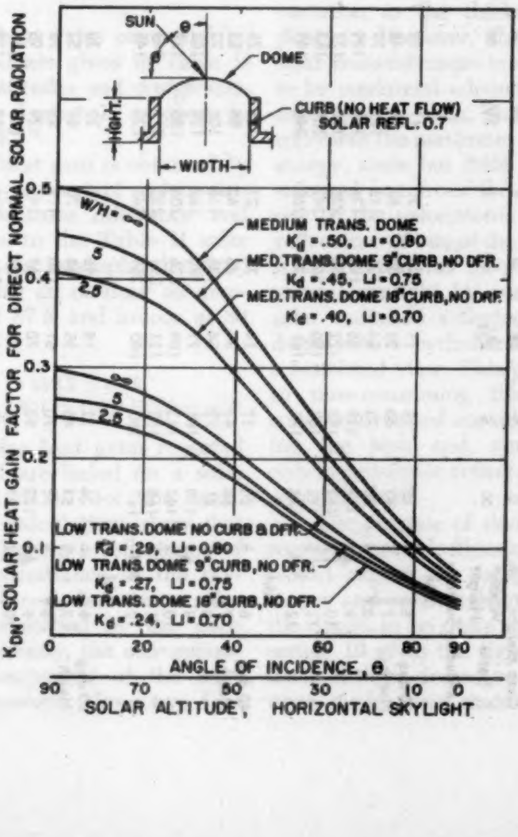


Table II—Solar Heat Gains through Horizontal Domed Plastics Skylights August 1, Clear Atmosphere[a] ($44\frac{1}{4}$ by $44\frac{1}{4}$-in. opening) Btu/(hr) (sq. ft of opening)

Skylight			Clear Dome & Light Diffuser			Clear Dome (No Diffuser)			Med. Trans. Dome		(No Dfr)	Low Trans Dome (No Dfr)			Flat Glass Reg. Plate[e]
Curb Height,[b, c] in.			0	9	18	0	9	18	0	9	18	0	9	18	0
Ratio Width[d]/Height			∞	5	2½	∞	5	2½	∞	5	2½	∞	5	2½	∞
Latitude	Suntime AM ↓	PM ↑													
30 deg North	6	6	15	12	11	26	20	17	19	17	15	11	10	9	16
	7	5	47	40	30	80	63	54	48	43	38	27	25	23	64
	8	4	80	73	54	136	116	101	77	70	62	44	40	37	124
	9	3	118	112	85	192	174	150	109	98	88	63	57	53	178
	10	2	152	143	122	238	220	198	137	123	111	79	73	66	220
	11	1	172	167	150	266	249	229	153	137	124	89	82	75	246
	12	PM	178	175	161	278	260	240	159	142	129	93	84	78	255
40 deg North	6	6	20	17	14	36	27	23	25	22	20	14	13	12	23
	7	5	50	43	34	85	68	58	51	45	40	29	26	24	69
	8	4	81	73	54	136	116	101	77	70	62	44	40	37	124
	9	3	110	101	77	180	162	139	102	91	82	59	53	49	169
	10	2	142	133	109	226	206	185	129	116	104	74	67	60	208
	11	1	160	152	133	251	232	211	144	130	117	84	76	70	234
	12	PM	167	160	140	259	242	220	149	134	120	86	79	72	243
50 deg North	6	6	26	22	19	46	35	30	30	26	23	17	16	14	31
	7	5	51	45	35	88	70	61	52	47	42	30	27	25	73
	8	4	78	70	51	130	111	95	75	68	60	43	38	36	117
	9	3	102	93	70	167	149	129	95	86	77	55	49	46	155
	10	2	127	116	93	202	184	165	115	103	93	66	60	55	188
	11	1	142	133	109	226	206	185	129	116	104	74	67	60	209
	12	PM	147	137	114	233	214	193	133	120	108	77	70	64	214
U-Volume[c]	UP		53	51	56	1.0	0.9	0.85	1.0	0.9	0.85	1.0	0.9	0.85	
Heat Flow	Down		0.46	0.43	0.40	0.8	0.75	0.7	0.8	0.75	0.7	0.8	0.75	0.7	

[a] ASHRAE GUIDE Values
[b] Curb reflectance 0.7 for solar radiation.
[c] Assumed, no heat flow through curb.
[d] Approximate ratio of width of opening to height of curb.
[e] Normal transmittance = 0.77

11 a.m., Aug. 1, 40 deg North Latitude, the 1960 GUIDE, Chapter 13, Table 6, gives the solar altitude as 64.5 deg. The direct normal radiation is about 286 Btu/hr/sq ft and the diffuse radiation falling on a horizontal surface is about 34.5. These values are from Table 4, Chapter 13, of the GUIDE. From Fig. 6 of this paper, the K_{DN} value from the curve is 0.475, K_d is 0.47 and U is 0.43. Let $t_o - t_i$ = zero. From equation 4

$$\text{Solar gain} = 0.475 \times 286 + 0.47 \times 34.5 + 43(0) = 136 + 16.2 = 152.2 \text{ Btu/hr/sq ft of opening}$$

The solar heat gains computed by this method are given in Table II for three latitudes and design conditions for August 1, as given in the GUIDE.

Total heat gain is obtained by adding the product of indoor-outdoor temperature difference and the U-value to the Table II solar gains. In our example the total heat gain for an outdoor air temperature of 87 F and indoor at 80 F is

$$\text{Total gain} = 152.2 + 0.43 (87 - 80) = 155.2 \text{ Btu/hr/sq ft}$$

The solar heat gains reported in Table II are based on a solar reflectance of 0.7 for the curb. Theoretical calculations show that for a curb having a reflectance of 0.4 the transmittance of the skylight is reduced, but more solar energy is absorbed by the curb and, consequently, the convection-radiation component of the heat gain is increased. These two factors counteract each other, and for the range of curb reflectance 0.4 to 0.7, the total solar heat gains are essentially unchanged. The exception to this is for a skylight consisting of a clear dome and curb which deviated more than 10% for incidence angles greater than 60 deg.

DISCUSSION

The method used for calculating the solar transmittance and reflectance of the domes does not include the energy which is lost through the edges and flanges of the dome, nor does it take into account the variation in the thickness of the plastics. However, the calculated solar transmittances are considered to be confirmed adequately by the experimental data, and probably represent the maximum transmitted energy, since but little radiation is reflected out from the black interior of the calorimeter. The solar reflectance of one of the translucent white domes was measured under natural sunlight by surveying the solar radiation reflected from the dome with a pyrheliometer having a restricted view. This process was so time-consuming that the sky conditions varied considerably during the test; and, consequently, only approximate reflectance values were obtained.

The purpose of skylights is, of course, to provide illumination. One would expect the light transmittances and solar transmittances of the domes to be quite similar. Reference 10 gives the light transmittance of three domes, two of which were made from material of the

same specifications as the low and medium transmittance domes reported here. By comparison the solar transmittance curves are similar in shape but are about 10 to 15 percentage points lower in magnitude. Also to be noted in this reference is that the light transmittance for the low transmittance dome is shown to be nearly diffuse but that the medium transmittance dome retains a little of its directional qualities.

The combined outside coefficient used for the design factors seems to fit the experimental data reasonably well. Variations of the outdoor coefficient have but a small effect upon the total heat gain for the dome with curbing and diffuser, and a somewhat larger effect upon heat gains for the dome alone. It has no effect upon the transmitted portion but only the convection-radiation gain. For the clear dome alone, a change in outdoor coefficient from 4 to 1 Btu/hr/sq ft/F will increase the K_{DN} value about 4% at normal incidence. For the low transmittance dome equivalent percentage is 15.

The over-all outdoor coefficient is composed of a convection exchange between the dome and the outdoor air and a radiation exchange between the dome and the sky or adjacent buildings.[3] With the dome at outdoor air temperature there would be no convection exchange with the outdoor air but it could lose heat to the sky by radiation, especially under clear sky conditions.[3] Thus a fixed combined outdoor coefficient only approximates the actual outdoor heat exchanges. In Fig. 13 of reference 1, the observed convection-radiation gain is given for flat glass skylights. This figure shows that, with no solar radiation falling on the flat-glass skylight, it would lose heat at the rate of 8 Btu/hr/sq ft for zero indoor-outdoor temperature difference. Therefore, assuming this figure to be correct, the maximum error in heat gain would be less than 8 Btu/hr/sq ft for the flat glass skylight given in Table II which is based on an outside coefficient of 4. Outside coefficients for flat roofs were reported[11] to vary from 1.14 to 3.82 Btu/hr/sq ft/F and an average value of 3.0 was used.

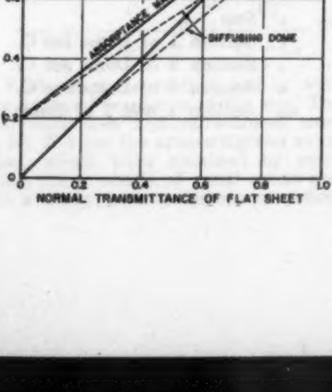
The U-values which are given in the paper are estimated values which check against observed data which were adjusted for an over-all outside coefficient of 4.0. Estimates were made assuming no heat flow through the curbing which, in effect, was quite low for the well insulated curbing used.

Solar heat gains for skylights other than those tested probably can be estimated from the theory. For a given dome transmittance, it is conceivable that all solar radiation not transmitted is either entirely reflected or entirely absorbed. The transmittance of a skylight is shown by calculation for a given flat sheet transmittance to be slightly higher for zero absorptance (maximum reflectance) than for zero reflectance (maximum absorption). The convection-radiation gain results from solar energy being absorbed by the dome and curbing, and reradiated or con-

vected to the space. For a single dome skylight without curb this amounts to 10 to 20% of the energy absorbed by the dome. The gain due to absorbed solar radiation never exceeded 12% of the solar radiation intensity for any of the skylights tested.

The probably maximum and minimum K-factors for clear and diffusing domes without curb or light diffuser are shown in Fig. 8. For a given transmittance the total solar heat gain is highest for a material having no solar reflectance, and lowest for a material having no solar absorptance. The skylights with curbing and with the diffuser follow the same trends percentage-wise. For diffusing type domes estimates can be made directly from Fig. 7. It should be pointed out that all of the data in this paper apply to a skylight having a square base. For this specific geometry the transmitted component of heat gain will be the same per unit area regardless of size. The convection-radiation component will vary slightly with size but for practical purposes this change can be neglected.

Fig. 8 Effect of solar transmittance, and limiting values of absorptance or reflectance on solar heat gain factors, zero incidence angle



CONCLUSION

The solar heat gains through a number of plastics dome skylights, which are typical of the types in use, can be determined for almost any location and time of the year by use of the design factors which are given in the paper and knowledge of the solar altitude and intensity. Solar heat gains are tabulated for three latitudes for the commonly accepted design day, August 1. Total heat gains for the skylight are determined by combining the solar heat gains with the U Δ T gains.

The solar heat gain factors are intended to serve as a guide for air-conditioning engineers to better evaluate the instantaneous loads.

ACKNOWLEDGMENT

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REFERENCES

1. ASHAE Research Report No. 1560—Heat Gains through Glass Skylight Fenestrations by D. J. Vild and G. V. Parmelee (ASHAE Transactions, Vol. 62, 1956, p. 89)

2. ASHVE Research Report No. 1333—Measurement of Solar Heat Transmission through Flat Glass by G. V. Parmelee, W. W. Aubele, and R. G. Huebscher (ASHVE Transactions, Vol. 54, 1948, p. 168)
3. ASHVE Research Report No. 1442 Radiant Energy Emission of Atmosphere and Ground, by G. V. Parmelee and W. W. Aubele (ASHVE Transactions, Vol. 58, 1952, p. 89).
4. Lighting Calculations for Thirty-Five Thousand Rooms by Philip F. O'Brien (Illuminating Engineer, April 1960, p. 215)
5. ASHAE Research Report No. 1528—Measurement of Angular Emissivity, by A. Umur, G. V. Parmelee and L. F. Schutrum (ASHVE Transactions, Vol. 61, 1955, p. 118)
6. The Two Radiometer Method for the Simultaneous Determination of Emissivity and Surface Temperature (Thermal Radiation Project, University of California, Report No. 6 Report Code NR-015-202 December 14, 1948, p. 9)
8. ASHVE Research Report No. 1267—Summer Weather Data and Sol-Air Temperature—Study of Data for New York City by C. O. Mackey and E. B. Watson (ASHVE Transactions, Vol. 51, 1945, p. 90)
9. ASHRAE Research Report No. 1696—Heat Flow Through Glass With Roller-Shades by Necati Ozisik and L. F. Schutrum (ASHRAE Transactions, Vol. 65, 1959, p. 714)
10. Efficiency of Domed Acrylic Skylights by Edward Linforth, (Illuminating Engineer, Oct. 1958, p. 544)
11. ASHAE Research Report No. 1545—Periodic Heat Flow Through Flat Roofs by D. J. Vild and M. L. Erickson (ASHVE Transactions, Vol. 61, 1955, p. 406)

NOMENCLATURE

A Solar absorptance of an element when in combination with other elements of skylight, dimensionless

C_1 Projection of the area of the base of dome for a given incidence angle (See Fig. B-1), area

C_2 Projected area of dome excluding the C_1 area

C_4 Area of dome in defilade from direct sunlight

C + R Convection and radiation gain by room Btu/hr/sq ft

F Shape factor, Fa, b means the portion of energy leaving a which is intercepted by b, dimensionless

$F_{s,\,CRB}$ Portion of direct sunlight passing through the dome which falls on the curb, dimensionless

h Thermal conductance, Btu/hr/sq ft/F

H Height of curb

I Intensity of solar radiation, Btu/hr/sq ft

K Solar heat transfer factor, dimensionless

$L_{b,a}$ Transfer function giving the total average emittance of surface b resulting from initial (before interreflection) average emittance of surface a, dimensionless

R Reflectance for solar radiation including interreflections for itself, dimensionless

R' Reflectance for solar radiation excluding interreflection

T Transmittance for solar radiation including interreflections from itself, dimensionless

T' Transmittance for solar radiation excluding interreflections, dimensionless

t Temperature (F)

S Cross sectional area of curbing or opening area of dome or diffuser, (sq ft)

U Over-all coefficient of heat transfer Btu/hr/sq ft/F

y Convection resistance, (F) (hr) (sq ft) per Btu

z Radiation resistance, (F) (hr) (sq ft) per Btu

Z Thermal resistance, (F) (hr) (sq ft) per Btu

θ Angle of incidence, degrees

ϕ Solar altitude, degrees

α Solar absorptance, dimensionless

Subscripts

DFR	Light diffuser
DM	Dome skylight
CRB	Curbing
c	Convection
D	Direct solar radiation
d	Diffuse or sky radiation
i	Inside
N	Normal to sun's rays
o	Outside
r	Radiation
s	Sun
1	Section 1 of Dome see C_1
2	Section 2 of Dome see C_2
3	Section 3 of Dome see C_3
1+2	Section 1 and 2 of Dome

APPENDIX A
SOLAR PROPERTIES OF FLAT PLASTICS SHEETS

Solar Properties of flat sheets as determined by pyrheliometer measurement and used with some slight adjustment, for the computations given in the paper, are shown in Table A-1.

Table A-1 Solar Properties of Flat Acrylic Plastics Sheets

(Normal Intensity of Direct Solar Radiation)

Sheet	Incidence Angle, Θ	0	30	60	75 Estimated
Clear 3/16 in.	Transmittance	0.86	0.74	0.39	0.14
	Reflectance	0.04	0.04	0.05	0.06
Med Trans 3/16 in.	Trans	0.52	0.45	0.23	0.08
	Refl.	0.36	0.31	0.21	0.14
Low Trans 3/16 in.	Trans	0.27	0.23	0.12	0.04
	Refl.	0.57	0.49	0.31	0.19

Values were selected from curves plotted from experiment points. For the clear plastics the transmittances were computed for incidence angles of 60 and 75 deg.

APPENDIX B
TRANSMITTANCE OF DOMES

Clear Dome The computed transmittance of the clear dome was based on the sunlight passing directly into the room through section 1 of Fig. B1 and the energy entering through section 2 which was diffusely reflected from 3. The reflections from 3 and in turn from sections 1 and 2 of the dome were treated as infinite in number. The transmittance based on incident solar radiation is:

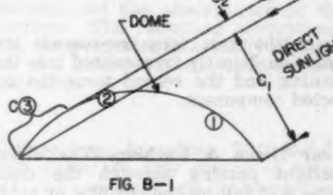

FIG. B—I

$$T_{(DM)} = \frac{\text{(Directly Transmitted Radiation) \& (Radiation Reflected to the Inside)}}{\text{Incident Radiation}} \tag{B-1}$$

$$T_{(DM)} = \frac{C_1}{C_1 + C_2} T'_1 + \frac{C_2}{C_1 + C_3} T'_2 R_3 \left(\frac{F_{3,0} + R_{2+1} F_{2+1,0} F_{3,2+1}}{1 - R_3 F_{3,2+1} R_{2+1} F_{2+1,3}} \right) \cdot \tag{B-2}$$

Each of these factors varies with the angle of incidence of the sun. The transmittances and reflectances used in Eq. B-2 are the area-weighted averages, which were obtained by summing the products of zonal areas and the average transmittances or reflectances of the zone over the section considered, and dividing it by the area of the section. The surface of the hemisphere and the segment were divided into zonal areas corresponding to increments of incidence angles of 15 deg.. The transmittances and

reflectances corresponding to the average incidence angle of each zone were selected from the data of the flat plastics sheets given in Appendix A. These area-weighted solar properties and the transmittance of the dome as given by Eq. B-2 were calculated for solar altitudes from zero to 90 deg.

Since area $C_1 + C_2$ is not readily known, the transmittance for solar radiation measured normal to the sun per unit area of the opening $(C_1 + C_2)_{base}$ is more easily used.

$$T_{(DM)N} = T_{(DM)} \frac{(C_1 + C_2)\,\Theta}{(C_1 + C_2)_{base}} \qquad \text{(B-3)}$$

Diffuse Transmitting Domes — The dome was assumed to transmit perfectly diffusely and all reflections also were taken as diffuse.

$$T_{(DM)} = T_{1+2}\,F_{1+2,0} + T_{1+2}\,F_{1+2,3} \left[\frac{R_3\,(F_{3,0} + F_{3,1+2}\,R_{1+2}\,F_{1+2,0})}{1 - F_{1+2,3}\,R_3\,F_{3,1+2}\,R_{1+2}}\right] \qquad \text{(B-4)}$$

where the first term represents the radiation directly transmitted into the opening and the second term the reflected component.

Clear Dome & Curbing—The direct sunlight passing through the clear dome will fall on the curbing or enter the room. The response-excitation transfer functions were used for these calculations and, for this method of approach, the portion of the solar radiation initially falling on the curbing and reflected from the curbing was assumed as a source of energy which is diffuse in nature. The equation expressing this relationship for direct normal solar radiation is:

$$T_{(DM,\,CRB)N} = T_{(DM)N}\,(1 - F_{S,CRB}) + T_{(DM)N}\,F_{S,CRB}\,R_{CRB}\,\frac{L_{room}}{\overline{CRB}}\,\frac{\alpha_{room}}{R_{room}}\,\frac{S_{DM}}{S_{CRB}} \qquad \text{(B-5)}$$

The first term in the equation is the energy directly transmitted through the dome into the room. The second term is the energy reflected from the curbing which eventually is absorbed by the room after multiple reflections. The transfer functions are based on perfectly diffuse reflections and a plane uniform source of energy. These assumptions only approximate the experimental conditions. The transfer function $L_{\frac{room}{CRB}}$ is the portion of solar radiation originally reflected from the curbing which is rereflected from the room to the skylight. In order that the value of the transfer function be finite, a room reflectance of 0.03 was used.

Diffuse Dome and Curbing—This dome is assumed to transmit only diffusely and thus the transmitted energy is related to the opening or room response from dome excitation.

$$T_{(DM,CRB)N} = T_{(DM)N}\,L_{\frac{room}{DM}} \times \frac{\alpha_{room}}{R_{room}} \qquad \text{(B-6)}$$

Clear-Dome, Curbing, and Light Diffuser—This treatment was similar to that of the clear dome and curbing with additional terms for the diffuser.

$$T_{(DM,CRB,DFR)N} = T_{(DM)N}\,(1 - F_{S,CRB})\,\frac{T_{(DFR)N}}{\mathrm{Cos}\,\Theta} + T_{(DM)N}\,\frac{T_{(DFR)d}}{R_{(DFR)d}} \left[L_{\frac{DFR}{CRB}}\,F_{S,CRB}\,R_{CRB}\,\frac{S_{DM}}{S_{CRB}} + (L_{\frac{DFR}{DFR}} - 1)\,(1 - F_{S,CRB})\,R_{(DFR)d}\right] \qquad \text{(B-7)}$$

APPENDIX C
REFLECTANCE OF DOMES

For clear domes the reflectance of the clear dome was in part a reflectance from portions 1 and 2, Fig. B-1, and a transmittance through area 3 to the outside of radiation coming in through 2, plus diffuse reflections which pass out through the dome.

The reflectance was taken as

$$R_{(DM)} = R'_{1+2} + \frac{C_1}{C_1 + C_2} T'_3 \left[T'_3 + \frac{R_3 (1 - F_{3,e}) (R_{2+1} F_{3+1,3} T'_3 + T'_{2+1})}{1 - R_3 (1 - F_{3,e}) R_{2+1} F_{3+1,3}} \right] \tag{C-1}$$

and for radiation measured normal to the sun's rays

$$R_{(DM)} = R_{(DM)} \frac{(C_1 + C_2)\,\theta}{(C_1 + C_2)_{base}} \tag{C-2}$$

For diffuse domes the reflectance both inside and outside of the dome as well as the transmittance was assumed perfectly diffuse.

$$R_{(DM)} = R'_{1+2} + \frac{T_{1+2} F_{1+2,3} (R_3 F_{3,2+1} T_{1+2} + T_3)}{1 - F_{1+2,3} R_3 F_{3,2+1} R_{1+2}} \tag{C-3}$$

APPENDIX D
ABSORPTANCE OF DOMES

For dome only the absorptance for solar radiation was obtained by subtracting the computed transmittance and reflectance from unity.

$$\alpha_{(DM)} = 1 - T_{(DM)} - R_{(DM)} \tag{D-1}$$

Clear dome & curbing The response due to excitation from the curbing (direct beam reflected from the curbing) was divided by its own reflectance which results in the total solar radiation falling on that surface. The effective absorptance is merely the product of all energy falling on the surface and the absorptance of that surface. The following absorptances are the percentages actually absorbed of the direct normal incident solar radiation.

Diffuse dome & curbing. With the diffuse dome the only source of energy for the response excitation concept is the dome itself.

Dome absorptance

$$A_{(DM)N} = \alpha_{(DM)N} + T_{(DM)N} F_{S,CRB} R_{CRB} \frac{S_{(DM)}}{S_{CRB}} L_{\frac{DM}{CRB}} \frac{\alpha_{(DM)6}}{R_{(DM)6}} \tag{D-2}$$

Curb absorptance

$$A_{(CRB)N} = T_{(DM)N} F_{S,CRB} \left[\alpha_{(CRB)} + (L_{\frac{CRB}{CRB}} - 1) R_{CRB} \frac{S_{DM}}{S_{CRB}} \frac{\alpha_{CRB}}{R_{CRB}} \right] \tag{D-3}$$

Dome absorptance

$$A_{(DM)N} = \alpha_{(DM)N} + T_{(DM)N} \left(L_{\frac{DM}{DM}} - 1\right) \frac{\alpha_{(DM)d}}{R_{(DM)d}} \quad \text{(D-4)}$$

Curb absorptance

$$A_{(CRB)N} = T_{(DM)N} L_{\frac{CRB}{DM}} \frac{\alpha_{(CRB)d}}{R_{(CRB)d}} \quad \text{(D-5)}$$

Clear Dome, Curb, and Light Diffuser. Two possible sources of energy exist with this combination. These are the reflections of the direct beam from the curb and from the light diffuser.

Dome absorption

$$A_{(DM)N} = \alpha_{(DM)N} + \frac{\alpha_{(DM)d}}{R_{(DM)d}} T_{(DM)N} \left[F_{S,CRB} R_{CRB} \frac{S_{DM}}{S_{CRB}} L_{\frac{(DM)}{CRB}} + (1 - F_{S,CRB}) R_{(DFR)d} L_{\frac{DM}{DFR}} \right] \quad \text{(D-6)}$$

Curb Absorption

$$A_{(CRB)N} = \alpha_{(CRB)N} T_{(DM)N} F_{S,CRB} + \frac{\alpha_{(CRB)d}}{R_{(CRB)d}} T_{(DM)N} \left[F_{S\,CRB} R_{CRB} \frac{S_{DM}}{S_{CRB}} \left(L_{\frac{CRB}{CRB}} - 1\right) + (1 - F_{S,CRB}) \times R_{(DFR)d} L_{\frac{CRB}{DFR}} \right] \quad \text{(D-7)}$$

Diffuser Absorption

$$A_{(DFR)N} = \alpha_{(DFR)N} T_{(DM)N} (1 - F_{S,CRB}) + \frac{\alpha_{(DFR)d}}{R_{(DFR)d}} \left[(1 - F_{S,CRB}) T_{(DM)N} R_{(DFR)d} \left(L_{\frac{DFR}{DFR}} - 1\right) + F_{S,CRB} R_{CRB} \frac{S_{DM}}{S_{CRB}} L_{\frac{DFR}{CRB}} \right] \quad \text{(D-8)}$$

APPENDIX E
CONVECTION-RADIATION GAIN

Single dome—Here the convection and radiation gains by the space were analyzed according to Fig. E-1 and the results compared with the experimental data. The solar energy absorbed by the dome $\alpha_{(DM)N} I_{DN}$ was obtained from the conversion of equation D-1 to give the absorption for direct normal radiation. For $t_o = t_i$, the convection and radiation gain to the inside is

$$C\&R = \frac{Zo}{Zo + Zi} \alpha_{(DM)N} I_{DN} \quad \text{(E-1)}$$

where the reciprocal of the thermal resistance is conduction

$$C\&R = U \left(\frac{\alpha_{(DM)N} I_{DN}}{h_o} \right) \quad \text{(E-2)}$$

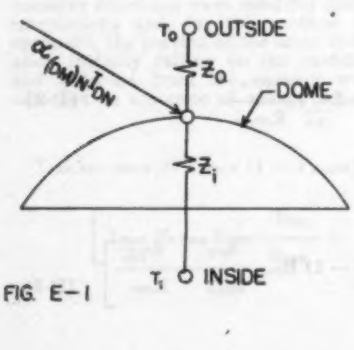

FIG. E-1

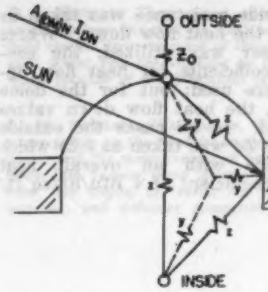

FIG. E–2

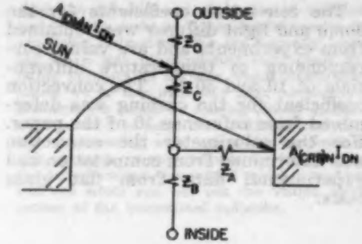

FIG. E–3

including the temperature effect

$$C\&R = U\left(\frac{\alpha_{(DM)N}\, I_{DN}}{h_o} + to - ti\right) \tag{E-3}$$

The U-value will vary with temperature difference and mean temperature and consequently is somewhat dependent upon the absorbed solâr energy. For general application the variation is neglected.

Dome and curbing. The thermal interchange resistance for the case of a dome and curb are shown in Fig. E-2. Three resistances marked z represent thermal radiation exchanges between the surfaces, and the three resistances y represent the convection exchange between surface and air. The curbing was assumed to be adiabatic for this analysis. The resistances marked z were first converted to a delta circuit, then combined with the delta circuit of the radiation resistances and the combination changed to the wye circuit shown in Fig. E-3. From this relation the connection and radiation gain to the inside is:

$$C\&R = \frac{1}{Zo + Zc + Zo}\,[Zo\, A_{(DM)N}\, I_{DN} + (Zo + Zc)\, A_{(CRB)N}\, I_{DN} + to - ti] \tag{E-4}$$

$$C\&R = U\left[\frac{A_{(DM)N}\, I_{DN}}{h_o} + (Zo + Zc)\, A_{(CRB)N}\, I_{DN} + t_o - t_i\right] \tag{E-5}$$

Dome, curbing, and light diffuser. The thermal circuit for this combination is similar to that shown in Fig. E-2 with the junction labeled "inside" becoming instead the diffuser. Also a thermal resistance must be added connecting the diffuser with the inside.

$$C\&R = \frac{1}{Zo + Zc + Z_B + Zi}\,[\,[Zo\, A_{(DM)N}\, I_{DN} + (Zo + Zc)\, A_{(CRB)N}\, I_{DN} + (Zo + Zc + Z_B)\, A_{(DFB)N}\, I_{DN} + t_o - t_i] \tag{E-6}$$

The range of convection and radiation coefficients shown in Table E-1 were selected for evaluation of the thermal circuits.

Table E-1 Radiation & Convection Coefficients

	Radiation Coef. h_r	Convection Coefficient h_c UP	Convection Coefficient h_c DOWN
Dome	0.7-1.0	0.72-0.90	0.15-0.17
Curb 9' x 18'	0.7-1.0	0.60-0.85	0.60-0.85
Light Diffuser	0.7-1.0	0.61-0.76	0.13-0.14
Calorimeter or Room	0.7-1.0	0.45-0.48	0.17-0.18

The convection coefficients for the dome and light diffuser were obtained from experiments and are values corresponding to temperature differentials of 10 and 30 F. The convection coefficient for the curbing was determined from reference 10 of the paper. For the calorimeter the convection was determined from computation and experimental data from flat glass tests.

The inside resistance was taken as unity for the heat flow down. Where the diffuser was utilized, the convection coefficients for heat flow upwards were used; but for the dome and curb, the heat flow down values were used. In all cases the outside resistance Zo was taken as 0.25 which corresponds with an overall heat transfer coefficient of 4 Btu/hr/sq ft/F.

DISCUSSION

O. L. Pierson, Philadelphia, Pa. (Written): A related investigation carried out in our Laboratories used five calorimeters, with blackened, melting ice as the heat sink. One calorimeter integrated the incident solar energy while the others collected the heat transmitted by plastics domes. These results showed, on the basis of Table II of the current work, a heat gain of 178 Btu/hr/ft² for the medium transmittance dome and 105 for the low transmittance dome. Comparable values from Table II at noon and 30 deg N latitude are 159 and 93, making the domes some 10% more efficient than in 1957. The difference is largely explained by the new data on Plexiglas contained in Table A-1 of the present work. Our control calorimeter was covered with 3/16 in. flat sheet to which we assigned a somewhat higher transmittance than shown in Table A-1. Our use of the low temperature heat sink also made accurate determinations of the reradiated and convected components of greater importance.

Acrylic plastics sheets are made in a range of thicknesses with a variety of translucent whites and colors available in each thickness. Forming of the domes thins the sheets in amounts dependent upon the height of the dome. Application of the data presented in this paper must be done with these factors in mind and the engineer must be certain the skylights used are of the types and materials tested.

The new data on the effects of skylight wells and of the addition of ceiling diffuser panels below the domes is a welcome addition to our engineering knowledge. Ceiling diffusers are frequently used but their effects on solar heat loads have not been measured previously.

E. P. Palmatier, Syracuse, N. Y.: In the actual application of this type of skylight, especially where you have cast a concrete curb tied into a concrete roof slab, some of the incident energy coming through the dome might not run off into the roof slab. Has this been considered? If anything, wouldn't it probably reduce the effectiveness of any heat gain?

C. M. Humphreys: It is pointed out in the paper that the values in Table II are based on the assumption of no heat transfer through the curb. The designer is cautioned that where the curb is poorly insulated, a correction may be necessary. The amount of this correction will depend upon the type and size of curbing used, and the insulation applied.

E. C. Miles, Pittsburgh, Pa.: Can you explain the 4% reflectance from a transparent sheet; it would seem that it should be 4% per surface instead of 4% per sheet (about 8% would be expected for the sheet).

Author Schutrum: The reflectances for solar radiation as reported in Table A-I are measured values as obtained by means of an Eppley pyrheliometer with natural sunlight. The errors encountered with extremely low reflectances are such that no high degree of accuracy can be expected. The authors agree with Mr. Mills that 8% reflectance for a transparent sheet would be more realistic.

B. Y. H. Liu, Minneapolis, Minn.: This question relates to the method used in measuring solar radiation transmitted through plastics. There is no problem in measuring this transmission if the plastics is clear, but since there is a diffused surface how do you measure this transmittance by pyrheliometers? At the University of Minnesota we have the same kind of measurements. It is difficult to measure this transmittance and it is influenced by the size of the sheet because the radiation transmitter is diffusing in all directions and it also depends on the location of the pyrheliometer from the sheet. By adjusting it at a different distance from the sheet, readings for greater distances could be obtained.

O. L. Pierson: The work with the pyrheliometer was done by using a rather large sheet, about 4 ft square and located rather close to the pyrheliometer so that you could cover a full 180 deg angle and thus have a chance to integrate the total transmittance of that flat sheet. If the pyrheliometer was set any distance away from the diffusing property of the plastics sheet, it would have a serious effect on the readings.

B. Y. H. Liu: Are there any other methods of making the reflectance from the sheet?

O. L. PIERSON: There are a number of different photometers that are used in the visible light range. However most of these cannot be used when you are working in the infrared range because of the difficulties of measuring.

C. W. PHILLIPS, Washington, D. C.: This paper points up the need for more information concerning the measurement of energy broken down into various wave lengths because it is what this begins to border on. As different materials and different geometric shapes are used, the diffusion and absorption of different wave lengths make an Eppley pyrheliometer an unreliable instrument. We are currently looking for a way of simulating solar loads for rating purposes on surfaces which means that unless we go into extremely high temperature filaments we would be dealing almost entirely with infrared in using dark radiating surfaces. We are uncertain as to reasonable or dependable amounts in our methods for interpreting the actual energy transmitted which a surface will see when you cut out the visible light section of the transmitted radiation.

No. 1760

Air Infiltration through Revolving Doors

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Member ASHRAE

Air infiltration through doors may cause discomfort in areas near the doors, and thus it is desirable to know the rate of infiltration for proper sizing of the heating and cooling equipment. A recent research (1) supplied comprehensive information on air infiltration through swinging doors; but, only limited data (2) are available on air infiltration rates through revolving doors. The purpose of the present investigation is to provide additional information on this subject. The research has been carried out at the ASHRAE Research Laboratory under the guidance of the former Research Advisory Committee on Heating and Air-Conditioning Loads. Experiments were made both under heating and cooling conditions. With the motor-driven revolving door, the door speed and temperature and pressure differences between indoors and outdoors were recognized as significant factors influencing the air-infiltration rate. With manually operated doors, however, additional information regarding a representative traffic pattern and the corresponding average door rpm was needed. The amount of air turbulence inside and outside, and the condition of the door seals would influence infiltration to a certain extent. The research was planned to determine the influence of all these variables on air infiltration.

TEST SETUP

A plan view of the test setup is shown in Fig. 1. A revolving door, 6 ft - 4 in. diam and 6 ft - 10 in. high was installed just inside a double door entrance to the Laboratory. When the swinging doors

L. F. Schutrum is Research Supervisor, C. M. Humphreys is Assistant Director of Research and J. T. Baker is a Research Engineer at the ASHRAE Research Laboratory. N. Ozisik is with the Oak Ridge National Laboratory.

This paper was prepared for presentation at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

were opened a typical revolving door entrance was available for testing. The door was motor driven, and its speed could be adjusted by means of a variable-pitch pulley. The door could also be operated manually.

A plywood partition was erected across the entrance back of the door, and all cracks in the test room were caulked to make the enclosure as nearly air tight as possible.

The exhaust system E served several functions. It provided a means of producing any desired pressure differential across the revolving door, it exhausted the infiltration air which, in a normal installation, would diffuse to other parts of the building, and a nozzle at the duct entrance provided a means of measuring the quantity of the exhausted air.

The recirculating system R also served several purposes. The air drawn from the test room was discharged through a heating coil, thus providing a means of controlling the temperature in the test room during the heating season. The tracer gas was injected into this system between the fan discharge and the heating coil. The fan handled sufficient air, and the supply and return ducts were so located, that the tracer gas was well distributed and the air temperature in the space was reasonably uniform.

A cooling system also was provided to permit a reduction of the air temperature in the test room during the summer tests. This system recirculated air within the space.

After all equipment was installed and all cracks had been caulked, the revolving door opening was sealed and tests were made

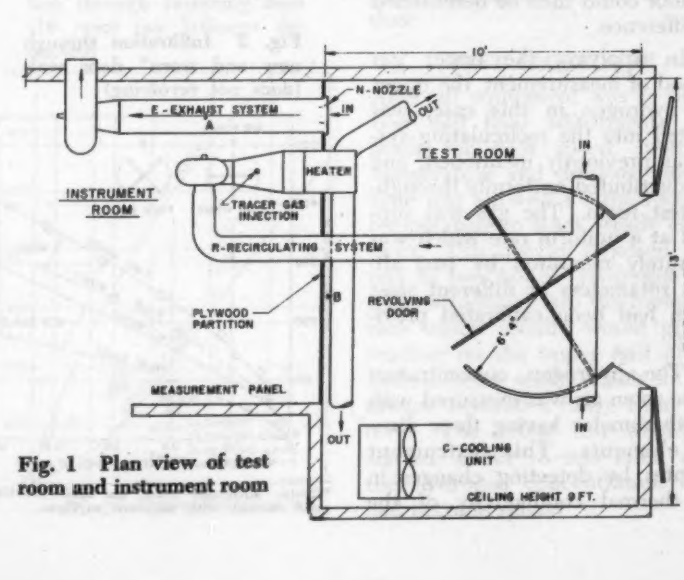

Fig. 1 Plan view of test room and instrument room

1760

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A cooling system also was provided to permit a reduction of the air temperature in the test room during the summer tests. This system recirculated air within the space.

After all equipment was installed and all cracks had been caulked, the revolving door opening was sealed and tests were made

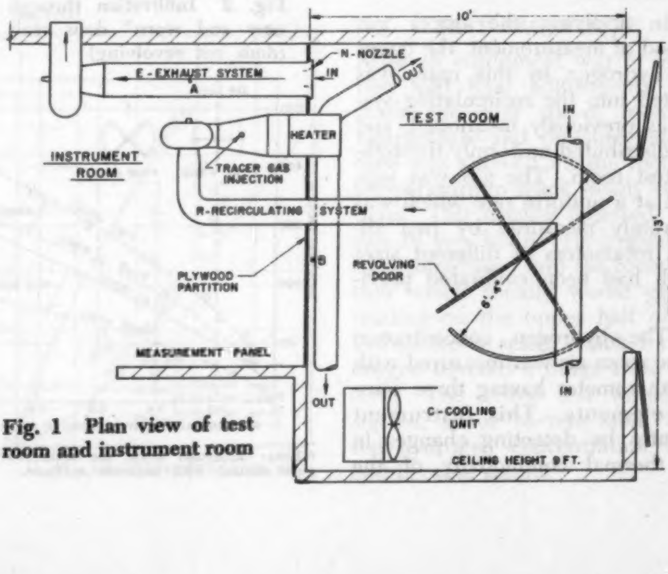


Fig. 1 Plan view of test room and instrument room

to determine the tightness of the test room at various pressure differentials. The uncontrolled leakage rate at a pressure differential of 0.5 in. of water was approximately 40 cfm.

Methods of measuring infiltration air – Outside air entered the test room by infiltration through the revolving door, and an equivalent amount left the space either through the exhaust system E or back out through the revolving door. These two routes are shown diagrammatically in Fig. 1-A of Appendix A. The problem was to devise measurement methods by which both of these flows could be determined. It was decided that this could best be accomplished by measuring the total infiltration by the tracer gas technique, and measuring the flow through the exhaust duct by means of a nozzle. The flow back through the door could then be determined by difference.

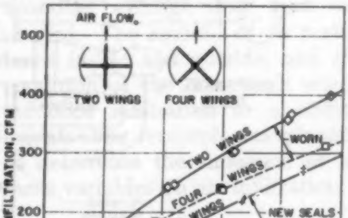
In applying the tracer gas method of measurement, the tracer gas, hydrogen in this case, was injected into the recirculating system as previously mentioned, and was distributed uniformly throughthe test room. The gas was supplied at a uniform rate which was accurately measured by two all-glass rotameters of different sizes which had been calibrated previously.

The hydrogen concentration in the room air was measured with a katharometer having three sensing elements. This instrument operates by detecting changes in the thermal conductivity of the mixture due to the presence of the tracer gas. The thermal conductivity of hydrogen is approximately seven times that of air, and the instrument is capable of detecting very small amounts of the tracer. The maximum concentration of hydrogen used in the tests was approximately 1%, which is well below the lower limit of flammability.

The scale of the katharometer was graduated to read in per cent of helium. It was therefore necessary to recalibrate the instrument for hydrogen. This was done by mixing known volumes of air and hydrogen in a large container, and recirculating the mixture through the sensing elements of the instrument.

A preliminary survey indicated that the concentration of hydrogen was quite uniform throughout the

Fig. 2 Infiltration through new and worn* door seals (door not revolving)

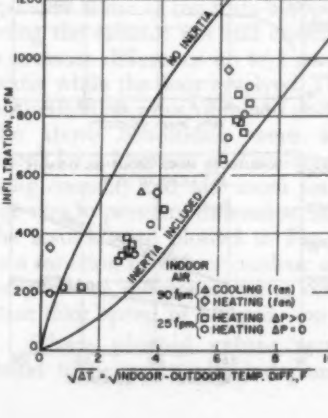


* Note: Although worn, the seals provided good contact with adjacent surfaces.

test room except in an area within about 3 ft of the revolving door opening, where the mixing of the room air and infiltration air was taking place. In all later tests, the hydrogen concentration was measured at the 72-in. level at point A in the test room and at point B in the exhaust duct. In tests with the exhaust fan operating, the concentration at both points was essentially the same (see Fig. 1).

Rapid diffusion of hydrogen into the air in the inside segment of the door would introduce errors in the results. However, estimates indicated that this diffusion rate was quite small. The method of calculating the infiltration rate from the tracer gas concentrations is explained in Appendix A.

Fig. 3 Observed data and calculated curves of infiltration through revolving door (10 rpm) (air leakages deducted)



Flow in the exhaust system was determined by a nozzle located at the duct inlet as shown in Fig. 1. The nozzle size was varied from 1 to 7 in., depending upon the rate of flow to be measured.

Other instrumentation—Differential pressures were measured with draft gauges. The outdoor pressure was obtained with a small static tube extending approximately 3 in. from the face of the wall.

The air temperature at several locations and elevations in the test room was sensed by copper-constantan thermocouples. These were connected into an indicating electronic type potentiometer.

A vane type anemometer located about 5 ft outside the door opening was used to measure the wind velocity. Air velocities in the test room were measured with a heated-thermocouple type anemometer about 3 ft away from the door.

TEST PROCEDURE

After all equipment was in operation and final adjustments were made for door speed, room air temperature and pressure, enough time was allowed for the room air temperature to reach steady conditions. This usually required from one to two hours. Hydrogen was then introduced into the room at a rate which usually would give a reading on the upper half of the katharometer scale, which for hydrogen covered a range from 0 to 0.7%. The actual test would be started, about 15 min later, after the hydrogen concentration level had become stable.

During the 15-min test period, air temperatures and pressures, and rotameter and katharometer readings were taken at fixed intervals. The wind velocity and the door speed also were recorded.

Tests were made under both heating and cooling conditions. Indoor-outdoor air temperature differences up to 60 F could be obtained in winter, while in summer the indoor temperature could be held as much as 30 F below the outside air.

The test program was planned to permit the evaluation of all of the important variables, including door speed, indoor-outdoor temperature and pressure differentials, indoor air motion and outdoor wind.

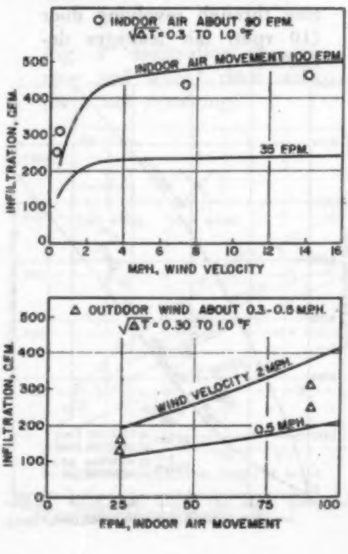
Analysis of air infiltration through revolving doors – For the purpose of analysis, air infiltrating through a revolving door can be separated into two components: One, component A, is the infiltration through the cracks between the door housing and door wings. The amount of this component depends upon the width and length of crack and the pressure difference between indoors and outdoors.

The other component, B, is the air infiltration related to door movement. When a segment of the revolving door filled with cold outdoor air is turned to the warm room side, circulation starts between the room and the segment air due to the density difference. A similar air exchange takes place at the opposite segment of the door with the outdoor air. The amount of this component of infiltration depends upon indoor-outdoor temperature difference, door speed and door size. Furthermore, indoor air motion and outdoor wind were expected to influence this infiltration. An analysis of this problem and the derivation of an analytical equation for calculating air infiltration related to the door movement is included within Appendix B.

TEST RESULTS

Infiltration Component A: Air leakage past door seals due to pressure differential – Infiltration past door seals depends upon the closeness of fit and the pressure differential.

Fig. 4 Effect of wind and indoor air movement on infiltration for small temperature differences and door speed of 5 rpm (air leakages deducted)



To investigate the magnitude of this infiltration, tests were made with two sets of new seals and one set of worn seals. The door was not revolving for these tests but was adjusted so that either two wings or four wings of the door were touching the door housing. Fig. 2 shows the magnitude of infiltration for pressure differentials up to 1 in. of water. These infiltration rates were corrected for room air leakage. With both new and used seals there were no visible cracks. Where visible, crack size can be estimated and the infiltration may be found from Fig. 10 of Reference 1. Data on pressure differentials for buildings can be found from the same reference.

Infiltration Component B: Air exchange related to door movement—Experiments were made with constant door speeds of 1, 2, 5 and 10 rpm, for various indoor-outdoor air temperature differences. The effect of indoor-outdoor pressure difference on the door movement component of infiltration was investigated in some of the tests by operating the exhaust fan and creating a pressure difference up to 1 in. of water while the door revolved. The air infiltration rates, obtained under the above conditions, were adjusted for infiltration past seals (two wing contact) and test room leakage due to pressure difference, and the results were plotted in Fig. 3 as a function of indoor-outdoor air temperature difference for a constant door speed of 10 rpm.

These plotted values were found to be practically independent of the pressure differential. This is shown in Fig. 3 by the proximity of the squares and circles which are the symbols for tests with and without pressure difference. The upper curve of Fig. 3 shows calculated infiltration rates neglecting the inertia of the air, and assuming still air conditions both indoors and outdoors. In calculating the lower curve the effect of inertia was included. This curve follows the observed data but lies below the points.

However, in the experiments, air motion indoors and wind outdoors were expected to increase the infiltration. Moreover, summer data showed higher infiltration rates than the winter data. The only difference in the test setup was the cooling blower used in the summer tests which caused higher indoor air velocities. The blower was then used for heating tests (coil not cooled) and the effect of air velocity on infiltration was verified. The experimental points in Fig. 3, representing the data with indoor air movement averaging about 90 fpm with the door stationary (triangles for summer and diamonds for winter data), are higher than those in which the average velocities were about 25 fpm (squares and circles).

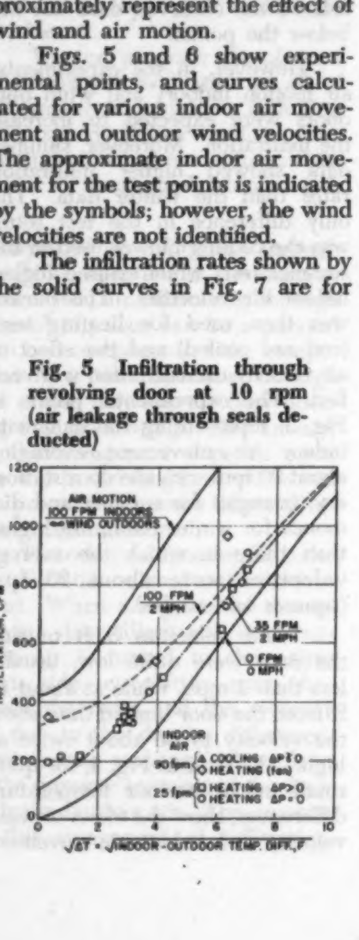
Wind velocities 5 ft outside the door were quite low, usually less than 2 mph, while at about 10 ft from the door limited data show the velocity to be about twice as high. The data in Fig. 4, for quite small indoor-outdoor temperature differences, show the effect of wind velocity and indoor air movement

on infiltration rates. The curves were calculated as described in Appendix B, based on the assumption that the heads due to the average non-directional air movement and wind velocities have the same effect on infiltration as equal heads due to temperature differences. The trends, if not precisely the magnitude, of the calculated curves in Fig. 4 are in agreement, and thus this premise was assumed to approximately represent the effect of wind and air motion.

Figs. 5 and 6 show experimental points, and curves calculated for various indoor air movement and outdoor wind velocities. The approximate indoor air movement for the test points is indicated by the symbols; however, the wind velocities are not identified.

The infiltration rates shown by the solid curves in Fig. 7 are for an indoor air movement of about 35 fpm and a wind velocity of 2 mph. The maximum wind velocity normal to the entrances at the curb line was reported by T. C. Min[1] to be about 2 mph when the wind velocities at the U. S. Weather Bureau were 20 mph. The dashed curves represent the infiltration with extremely high wind velocities which might cause one hundred per cent air exchange in the outside segment of the door.

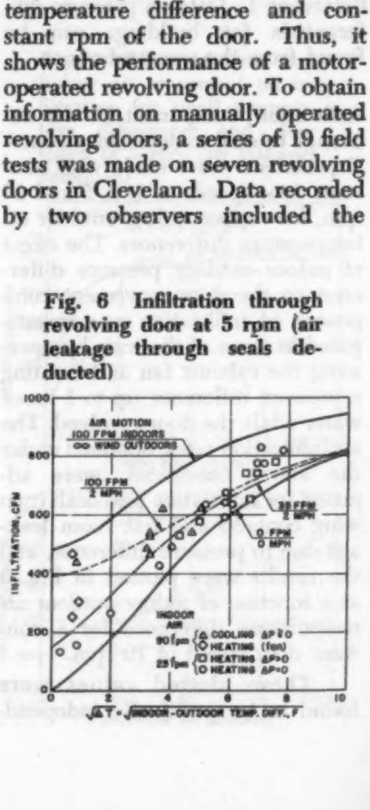
Fig. 5 Infiltration through revolving door at 10 rpm (air leakage through seals deducted)



FIELD TESTS

Fig. 7 shows the infiltration which may be expected as a function of temperature difference and constant rpm of the door. Thus, it shows the performance of a motor-operated revolving door. To obtain information on manually operated revolving doors, a series of 19 field tests was made on seven revolving doors in Cleveland. Data recorded by two observers included the

Fig. 6 Infiltration through revolving door at 5 rpm (air leakage through seals deducted)



elapsed stop watch time for a given number of door revolutions (50 or 100), time during this period when the door was idle, and a count of the number of people passing through in each direction. The traffic rates varied from 160 to 1770 people per hr. The two curves of regression in Fig. 8 show the relationship found from the field tests between traffic rate, and average rpm and operating time.

Infiltration (Component B) through manually operated and motor-driven revolving doors – The infiltration through manually operated doors for a given traffic rate was taken as the product of the operating time from Fig. 8 and the infiltration from Fig. 7 determined for the average rpm given in Fig. 8. To substantiate this method, three tests were conducted at the Laboratory with summer cooling conditions and with Laboratory personnel passing through the revolving door, manually operated, at traffic rates ranging from 1075 to 1975 people per hr. While this random traffic pattern was maintained, the tracer gas was being used to measure the infiltration. The observed infiltration, 400-450 cfm, was lower by about 10% on the average than that predicted.

The infiltration through manually operated revolving doors is shown in Fig. 9. The wind velocity was taken as 2 mph, but a few dashed curves for maximum wind velocity are given to show the probable upper limit of infiltration.

For motor-driven doors the in-

Fig. 7 Infiltration vs. rpm and indoor-outdoor air temperature difference (air leakage past seals deducted)

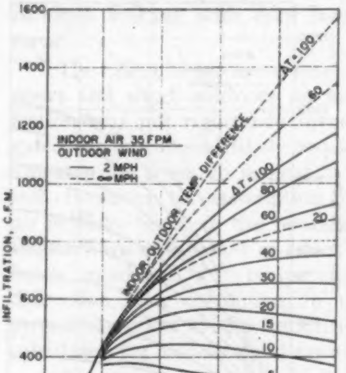
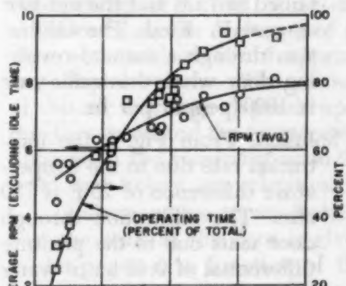


Fig. 8 Operating time and average rpm vs. traffic rate for seven manually operated revolving doors



filtration rates can be found from Fig. 7.

EFFECT OF PEOPLE

In a series of tests the doors were motor operated with none, one, two or four of the door segments occupied by people walking around continuously without leaving the space. The infiltration under these conditions was found to be lower by approximately the volume of the people occupying the space. However, when a person steps into the door space, he displaces part of the air in the space, and when he leaves the space on the opposite side of the door, an equal volume of air must replace him. This effectively increases the infiltration, counteracting the decrease caused by the occupancy of the door space. Calculations show that the effect of people on infiltration is small and can be neglected.

Example of infiltration calculation—

Given: A building has a pressure differential of 0.46 in. of water when the indoor air is maintained at 75 F and the outdoor at zero F. Find: The infiltration through a manual revolving door when the traffic rate is 1000 people per hr.

Solution: From Fig. 9 the infiltration rate due to the temperature difference of 75 F is 750 cfm. The infiltration through door seals due to the pressure differential of 0.46 in. of water is 250 cfm from Fig. 2 for two wings and worn seals. The total infiltration is 750 + 250 or 1000 cfm.

Second Example—

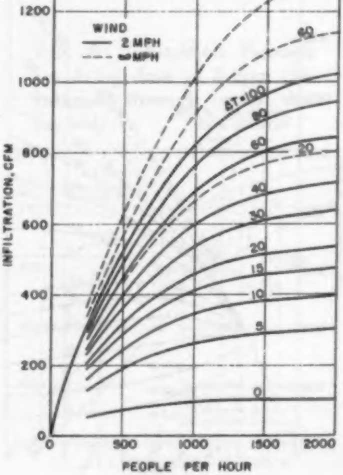
Given: The same conditions as in the previous example except that the door is motor driven and operates at a constant speed of 10 rpm.

Solution: The infiltration due to a temperature difference of 75 F is 1040 cfm as taken from Fig. 7. The infiltration due to a pressure difference of 0.46 in. of water is the same as in the first example, making the total 1040 + 250 or 1290 cfm.

DISCUSSION

All calculated infiltration curves and experimental data were based

Fig. 9 Infiltration through manually operated revolving doors (air movement 35 fpm indoors, air leakage past door seals deducted)



on the standard air density of 0.075 lb per cu ft. The calculated curves were not adjusted for the change in volume of the infiltration air due to temperature change as it passes from the outdoors to the indoors. Moreover, the infiltration rates found in the various figures which are a function of temperature difference were based on winter conditions. The same temperature differences for summer conditions would correspond to slightly lower density differences. It was felt that the refinements for these changes in air density were not necessary and would only complicate the presentation of the results.

One would expect that the motion of the door would tend to increase the crack area between door seals and the adjacent surfaces and thus increase the air leakage past the door seals due to pressure difference. The data showed only insignificant changes in crack leakage with door movement.

The effect of air movement indoors and wind outdoors on infiltration was not rigorously investigated, as this would likely require an extensive research project in itself. However, the assumption that air velocity heads in the boundary spaces were equivalent in effect to heads produced by temperature difference is a correction in the proper direction to the infiltration calculated for still-air conditions.

The centrifugal force of the air in the door segment was estimated from the difference in density of the incoming and outgoing air of the segment and the door speed. This effect was neglected in the final analysis since the magnitude of the force even at the higher rpm (10) was less than 10% of the temperature head.

In a few tests, heated air was introduced into the inside segment of the revolving door, through the opening provided for a luminaire in the ceiling of the revolving door housing. The rate of air entering through this opening was about 250 cfm. With but extremely limited data, a definite conclusion cannot be drawn; however, the infiltration rates deduced from these tests at 10 rpm were close to the values given by the upper curve of Fig. 5.

CONCLUSIONS

Infiltration through a revolving door can be estimated by combining the air leakage past the door seals with the infiltration related to the revolving of the door. The magnitude of air leakage through the seals of the door is the result of the pressure differential across the building entrance and the size of the openings at the seal. Revolving the door causes an exchange of indoor and outdoor air of approximate equal volume. The amount of this air exchange depends upon the door speed and the temperature differential and somewhat upon the wind and indoor air velocities.

Infiltration due to air leakage past the seals of the door is given in the paper as a function of the indoor-outdoor pressure differential. Infiltration related to the door movement is also given for a motor-driven revolving door, and for a

manually operated door for traffic rates up to 2000 people per hr.

Data given in this paper are based on the use of door seals which provide good contact with the adjacent surfaces. If seals deteriorate to the point that good contact is not maintained, leakage past the seals will increase greatly.

ACKNOWLEDGMENT

The authors recognize the valuable guidance of Messrs. A. M. Simpson and R. G. Chapman in setting up the research program, and also wish to thank the Revolving Door Div of the International Steel Door Company for its cooperation in supplying the revolving door for these experiments. Helpful suggestions from the staff of the Laboratory are acknowledged gratefully.

REFERENCES

1. ASHAE Research Report No. 1643, Winter Infiltration Through Swinging-Door Entrances in Multi-Story Buildings, by T. C. Min (ASHAE Transactions, Vol. 64, 1958, p. 421)

2. Infiltration Problem of Multiple Entrances by A. M. Simpson and K. B. Atkinson (ASHVE Journal Section, Heating, Piping & Air Conditioning, Vol. 8, No. 6, June, 1936, p. 345-351)

APPENDIX A

Infiltration rates from tracer gas measurement

The hydrogen and air flow-circuits in the test room are shown in Fig. 1-A. The hydrogen flow paths are designated as Q_{HT} the total steady rate of hydrogen introduced into the room, Q_{HD} the rate of hydrogen flowing out of the room through the duct, and Q_{HO} the net rate of hydrogen passing out through the revolving door. The air entering the test chamber is Q_{AI} through the revolving door, and Q_L the leakage through walls, etc. Air leaves the chamber via the revolving door Q_{AO} and the exhaust duct Q_{AD}. C_s and C_D are the concentrations of hydrogen in the space and in the exhaust duct.

By volume balances on

hydrogen $Q_{HT} = Q_{HO} + Q_{HD}$ A-1

and on air $Q_{AI} = Q_{AO} + Q_{AD} - Q_L$ A-2

The measured concentrations are

$$\text{in space } C_s = \frac{Q_{HO}}{Q_{AO} + Q_{HO}} \quad \text{A-3}$$

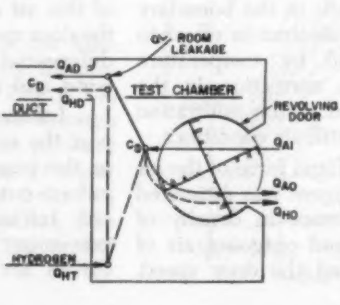


Fig. 1A

where from Eq. A-3 and A-1

$$Q_{AO} = \left(\frac{1 - C_s}{C_s}\right) Q_{HO} = \left(\frac{1 - C_s}{C_s}\right) (Q_{HT} - Q_{HD}) \qquad \text{A-4}$$

and in duct

$$C_D = \frac{Q_{HD}}{Q_{AD} + Q_{HD}} = \frac{Q_{HD}}{Q_D} \qquad \text{A-5}$$

from which

$$Q_{HD} = C_D\, Q_D \qquad \text{A-6}$$

from A-5 and A-6

$$Q_{AD} = (1 - C_D)\, \frac{Q_{HD}}{C_D} = (1 - C_D)\, Q_D \qquad \text{A-7}$$

Substituting Eq. A-6 into A-4

$$Q_{AO} = \left(\frac{1 - C_s}{C_s}\right) (Q_{HT} - C_D\, Q_D) \qquad \text{A-8}$$

and substituting Eq. A-7 and A-8 into A-2

$$Q_{AI} = \left(\frac{1 - C_s}{C_s}\right) (Q_{HT} - C_D\, Q_D) + Q_D\, (1 - C_D) - Q_L \qquad \text{A-9}$$

The infiltration was determined from Eq. A-9 for observed values of C_s and C_D from katharometer measurements, Q_{HT} from rotometer readings, and Q_D from nozzle information. Q_L was determined by measurement and

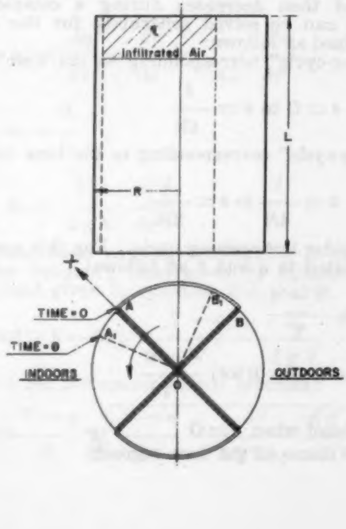


Fig. 1B

its relation to indoor-outdoor pressure differences. For tests without pressure differences the exhaust volume Q_D was zero and Q_L was assumed to be zero.

APPENDIX B

Calculation of air infiltration component B through a revolving door

Let a revolving door having 4 segments rotate at a uniform speed, N, in the direction shown in Fig. 1-B. Assume outside air is colder than room air.

Consider a quarter-section OAB having its leading edge OA along ox reference axis $\theta = O$. At this moment assume that air in the segment has a uniform temperature between indoor and outdoor air temperatures. At time θ let the segment be at position OA_1, and the volume of warm air entering into the segment during the time interval θ be q. Assume air displacement is taking place as the result of buoyancy due to cold air in the segment and warm air in the room, and that warm air entering the segment is collecting in a uniform layer at the top of the segment. At the time θ, the rate of change of q can be expressed as follows:

$$\frac{dq}{d\theta} = CA\sqrt{2gh} \qquad \text{B-1}$$

where

A = area of opening
h = head of air
C = flow coefficient

In this equation, both A and h are functions of θ and q. Therefore, in order to solve equation B-1, A and h are to be expressed in terms of q and θ. As area, A, first increases and then decreases during a complete cycle, the equation can be solved separately for the two time intervals defined as follows:

1. "Opening-cycle" corresponding to the time interval:

$$\theta = O \text{ to } \theta = \frac{1}{4N}$$

2. "Closing-cycle" corresponding to the time interval:

$$\theta = \frac{1}{4N} \text{ to } \theta = \frac{1}{2N}$$

First, consider the opening cycle. For this cycle, h and A can be related to q and θ as follows:

$$h = h_o \frac{V - q}{V} \qquad \text{B-2}$$

$$A = \tfrac{1}{2} L\,(2\pi RN\theta)\,\frac{V - q}{V} \qquad \text{B-3}$$

where

h_o = head when $\theta = O$
V = Volume of the door segment

Substituting B-2 and B-3 in B-1:

$$\frac{dq}{d\theta} = CL\,\pi\,RN\,\theta\frac{V-q}{V}\sqrt{2\,gh_o\frac{V-q}{V}} \qquad \text{B-4}$$

During the closing-cycle the time interval is $\frac{1}{4N} < \theta \leqq \frac{1}{2N}$.

For convenience take $\frac{1}{4N}$ as zero θ for reference purpose, thus the value of A for this closing cycle is

$$A = \tfrac{1}{2}\,L\,2\,\pi\,RN\left(\frac{1}{4N} - \theta\right)\frac{V-q}{V} \qquad \text{B-5}$$

and equation B-4 becomes

$$\frac{dq}{d\theta} = CL\,\pi\,RN\left(\frac{1}{4N} - \theta\right)\frac{V-q}{V}\sqrt{2\,gh_o\frac{V-q}{V}} \qquad \text{B-6}$$

Inertia Effect—In the foregoing derivation the inertia of air in the segment, that is, the head lost in accelerating this mass of air from initially zero to some finite velocity, was not considered. In a door segment, the inertia head at a time θ can be approximated as follows:

$$h_i = \frac{1}{2}\,\frac{L}{g}\,\frac{dv}{d\theta} \qquad \text{B-7}$$

where L/2 is used to be consistent with h_o, which is described later

since

$$vA_s = \frac{dq}{d\theta} \text{ or } \frac{dv}{d\theta} = \frac{1}{A_s}\,\frac{d^2q}{d\theta^2}$$

and

$$A_s = \frac{V}{L}$$

then

$$h_i = \frac{L^2}{2\,g\,V}\,\frac{d^2q}{d\theta^2} \qquad \text{B-8}$$

and the actual head causing infiltration at any time θ would be the buoyancy head given by equation B-2 less the inertia head given by equation B-8, that is

$$\text{Actual head} = h - h_i = h_o\frac{V-q}{V} - \frac{L^2}{2\,g\,V}\,\frac{d^2q}{d\theta^2} \qquad \text{B-9}$$

Equation E-4 for the opening cycle becomes:

$$\frac{dq}{d\theta} = K\theta\,\frac{V-q}{V}\sqrt{\frac{V-q}{V} - \frac{L^2}{2\,gV\,h_o}\,\frac{d^2q}{d\theta^2}} \qquad \text{B-10}$$

and for the closing cycle

$$\frac{dq}{d\theta} = K\left(\frac{1}{4N} - \theta\right)\frac{V-q}{V}\sqrt{\frac{V-q}{V} - \frac{L^2}{2\,gV\,h_o}\frac{d^2q}{d\theta^2}}$$

where

$$K = CL\,\pi\,RN\sqrt{2\,gh_o}$$

B-11

The head at the beginning of the opening cycle is

$$h_o = \tfrac{1}{2}\,L\,\frac{\rho_{oo} - \rho_1}{\rho} \qquad \text{B-12}$$

where

ρ_{oo} = density of air in the door segment at the beginning of the opening cycle

ρ_1 = density of air in room

ρ = reference density of air 0.075.

Fig. 2-B shows the air exchange rates for all four segments as calculated by equations B-10 and B-11 for assumed values of air temperature in the segment and indoors or outdoors. Note it does not represent the net infiltration but merely the amount of air entering or leaving the door segments.

In Equation 12 the determination of ρ_{oo}, the density of the air in the door segment, for a given indoor-outdoor temperature difference, requires a knowledge of the air temperature in the space. Thus in Fig. 3-B, t_{oc} is the air-space temperature just after closing. By inspection of the figure, equations can be written as

$$t_{oo} = f_o\,t_o + (1 - f_o)\,t_{oc} \qquad \text{B-13}$$

and

$$t_{oc} = f_i\,t_i + (1 - f_i)\,t_{oo} \qquad \text{B-14}$$

where

f_o = (Air displacement as fraction of segment volume when segment is exposed to outdoors) = $\frac{q_o + q_c}{V}$ outdoors B-15

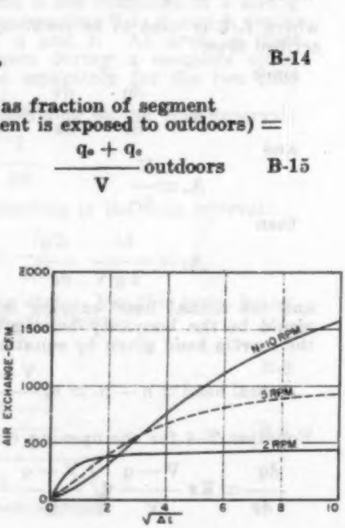


Fig. 2B Air exchange from one side of revolving door into door segment (includes inertia effect)

f_i = (Air displacement as fraction of segment volume when segment is exposed to indoors) =

$$\frac{q_o + q_c}{V} \text{ indoors} \qquad \text{B-16}$$

Combining equations B-13 and B-14

$$t_{sc} = \frac{f_i\, t_i + (1 - f_i)\, f_o\, t_o}{1 - (1 - f_o)\,(1 - f_i)} \qquad \text{B-17}$$

and

$$t_{so} = \frac{f_o\, t_o + (1 - f_o)\, f_i\, t_i}{1 - (1 - f_o)\,(1 - f_i)} \qquad \text{B-18}$$

The temperature of Segment I depends upon the relative quantity of indoor air m_I, and outdoor air, m_o in the segment.

$$t_{so} = \frac{t_i\, m_I + t_o\, m_o}{m_I + m_o} = \frac{t_i\, m_I + t_o\, m_o}{V \text{ (volume of segment)}} \qquad \text{B-19}$$

and similarly

$$t_{sc} = \frac{t_i\, m_{II} + t_o\, m'_o}{m_{II} + m'_o} = \frac{t_i\, m_{II} + t_o\, m'_o}{V} \qquad \text{B-20}$$

From Equations B-18 and B-19 the amount of indoor air in Segment I is

$$m_I = \frac{f_i\,(1 - f_o)\, V}{1 - (1 - f_o)\,(1 - f_i)} \qquad \text{B-21}$$

where for convenience, t_o was assumed to be zero. Similarly from Equations B-17 and B-20 the amount of indoor air in Segment II is

$$m_{II} = \frac{f_i\, V}{1 - (1 - f_o)\,(1 - f_i)} \qquad \text{B-22}$$

The difference in the quantity of indoor air in the outgoing and incoming segments is the net infiltration. Hence,

Infiltration per segment per revolution $= m_{II} - m_I = \dfrac{f_o\, f_i\, V}{1 - (1 - f_o)\,(1 - f_i)}$ and the infiltration rate through

B-23

Fig. 3B

the door becomes

$$Q = \frac{4\,N\,f_i\,f_o\,V}{1 - (1 - f_o)\,(1 - f_i)}, \text{cfm} \qquad \text{B-24}$$

In addition to knowing the infiltration through the revolving door the corresponding indoor-outdoor temperature difference must be found. By subtracting both sides of Equation B-18 from t_i it becomes

$$(t_i - t_{io}) = \frac{f_o}{1 - (1 - f_o)\,(1 - f_i)}\,(t_i - t_o) \qquad \text{B-25}$$

Also by subtracting t_o from both sides of Equation B-17 and dividing this result by Equation B-25, an equation is found which may help in visualizing some of the temperature and infiltration relationships.

$$\frac{t_{oo} - t_o}{t_i - t_{io}} = \frac{f_i}{f_o} \qquad \text{B-26}$$

Fig. 2-B shows air displacement rates for the entire door (i.e. 4 segments) and for (N) revolutions of the door as a function of ΔT. For a given ΔT the (f) values can be determined from Fig. 2-B with the following relationship:

$$f = \frac{\text{cfm from Fig. 2-B}}{4\,V\,N} \qquad \text{B-27}$$

where 4V was taken as 218 cu ft for the revolving door used in the tests.

For a given value of $t_i - t_{io}$, the net infiltration rate Q. and the corresponding indoor-outdoor temperature difference $(t_i - t_o)$ can be determined as follows:

1. Assume $t_i - t_{io}$ and obtain cfm (inside) from Fig. 2-B and determine f_i from Equation B-27.
2. Assume f_o. Where buoyance effects alone determine the infiltration, it is logical that $f_o = f_i$.
3. Knowing f_i and f_o, determine infiltration from Equation B-24 and $t_i - t_o$ from B-25.

Effect of turbulence and wind—By definition, the factors f_o and f_i represent the fractional air displacement of the segment volume when the segment is exposed to outdoors and indoors, respectively. This air displacement is caused by the temperature difference, in the absence of any external air turbulence or wind. Any external air movement will increase the value of f_o and/or f_i.

An approximate solution to the effect of air movement on infiltration was made by assuming that the average nondirectional air velocity head $\frac{v^2}{2g}$, was equivalent in its effect on infiltration to a buoyancy head of equal magnitude. For convenience, since curves of infiltration rates were plotted against the square root of the temperature difference between door segment air and room air, Fig. 2-B, the velocity heads were converted to temperature difference heads. Thus ΔT_T and ΔT_W were designated indoor turbulence and outdoor wind equivalent temperature heads.

To solve for the infiltration through the revolving door with a turbulence head inside of ΔT_T and wind outdoor equivalent to ΔT_W a trial and error solution was used as follows:

1. Assume $(t_i - t_{ee})$ and ΔT_T.

2. Find cfm_i and thus (f_i) from Fig. 2-B for $\sqrt{(t_i - t_{ee}) + \Delta T_T}$

3. Assume f_o.

4. Find $\sqrt{(t_{ee} - t_o) + \Delta T_W}$ from Fig. 2-B for assumed (f_o).

5. Calculate $(t_{ee} - t_o)$ from Equation B-26 if $[(t_{ee} - t_o)$ from Step 5$] > [(t_{ee} - t_o + \Delta T_W)$ from Step 4$]$ then a new value of (f_o) in Step 3 is required.

6. Solve for $\Delta T_W = [(t_{ee} - t_o) + \Delta T_W] - [t_{ee} - t_o]$.

7. Calculate $(t_i - t_o)$ from Equation B-25.

8. Calculate infiltration from Equation B-24.

From this procedure the indoor-outdoor temperature differential was found in Step 7, (reference $t_o =$ zero) for the infiltration rate determined in Step 8, turbulence indoors of ΔT_T and wind outdoors of ΔT_W. In this manner the curves of Fig. 7 were constructed.

DISCUSSION

C. KUMMER, Cincinnati, Ohio: How do these values compare with the GUIDE, which now states that you may take 50% of single doors when you have a revolving door.

AUTHOR HUMPHREYS: I am not sure just what recommendation the latest GUIDE makes regarding revolving doors, and can not answer your question. However, it seems unlikely that such a relationship would exist.

B. Y. H. LIU, Minneapolis, Minn.: What are the advantages of using the hydrogen gas as a tracer gas as compared to other gases, such as helium?

AUTHOR HUMPHREYS: So far as we know, there is no advantage in using hydrogen instead of helium as a tracer gas, and the explosive hazard associated with hydrogen is a distinct disadvantage. The katharometer we used was actually calibrated for helium. However, when the project was started helium was unavailable, and the only other suitable gas which could be obtained was hydrogen. Both helium and hydrogen have thermal conductivities quite different from air. In all tests the hydrogen concentration maintained was well below the explosive limit.

T. C. MIN, Auburn, Ala. (Written): Having been associated with the problems on infiltration prior to the period of the investigation reported in this paper, perhaps I am in a better position than some to express my appreciation to Mr. Schutrum and his associates of the Research Laboratory for the information presented. I would like to extend my congratulations to the Laboratory Staff for this important work without which the study of entrance infiltration cannot be considered to be completed.

It is interesting to note that from the example of the paper, the infiltration rate through a revolving door under the representative field conditions (0.46 in. of water of pressure differential, 0 F outdoor, and 75 F indoor temperature) is 60 cu ft per person for a manually revolving door (at 1,000 persons per hour) and 32 cu ft per person for a motor-driven type (at 10 rpm). These are comparable to the GUIDE values of 75 cu ft per person for infrequent usage, and 40 cu ft per person for heavy usage.

In another paper* dealing with swinging-door entrances, under the same conditions the infiltration is about 550 cu ft per person for a vestibule-type entrance and 900 cu ft per person for a single-bank entrance. They are in contrast with the corresponding GUIDE values of 20 to 100 cu ft per person. These facts detract in no way from the merits of the authors' work. I am merely trying to point out the relative magnitude of infiltration through a revolving door entrance and a swinging-door entrance, and the comparison of values with those in the ASHRAE GUIDE.

It is also pleasing to see that the complete appendices have been published with the text, for without them complete analysis of the paper is impossible. Furthermore, they are not only valuable to other investigators

* "Winter Infiltration Through Swinging-Door Entrances in Multi-Story Buildings" by T. C. Min, ASHRAE Transactions, Vol. 64, 1958, p. 421.

but also exhibit the complexity of the problem and the skill and ingenuity in conducting the research, and in analyzing and interpreting the data. There are a few points in the Appendices about which I would like to question the authors:

1. In equations A-2 and B-19, should the mass instead of volume be considered since the specific volumes of outdoor air at 0 F and that of indoor air at 75 F are quite different?

2. In equation B-1, a flow coefficient C and area of opening A are introduced, where A varies with q and θ. I would appreciate knowing the way that C is determined.

3. The establishment of the expression $vA_s = dq/d\theta$ is not all clear to me. What is the direction of velocity v, and why does it physically relate to A_s (the fan shape cross-section of the door segment) and $dq/d\theta$ (the time rate of the change of air)?

4. What are the solutions of the nonlinear differential equations B-10 and B-11? Have they been employed in constructing any of the graphs?

The answers to these questions will be appreciated.

AUTHOR SCHUTRUM: On behalf of the authors, I should like to thank Professor Min for his comments and particularly his comparison of the data in the GUIDE on infiltration through swing-door and revolving door entrances with the data obtained by the ASHRAE Laboratory on this subject. In answer to the questions which he raised on the appendices we offer this explanation.

The volume instead of the mass of the air was used in Equation A-2 merely because the katharometer was calibrated in terms of volume. Adjustments were made to correct the experimental data to a standard air density of 0.075 pounds per cu ft.

A basic assumption in these theoretical calculations of air-infiltration is that a uniform air temperature exists in the door segments I and II (Fig. 3B), that is complete mixing of incoming air and the remaining segment air is assumed. The volumes Ml and Mo in Equation B-19 are the volumes of the segment air and the incoming outside air after complete mixing occurs, both being at the same temperature.

The flow coefficient used in equation B-1 was taken as 0.6. No specific experimental verification was made except that the calculated infiltration approximated the measured rates.

The expression $\nu \, As = \frac{dq}{d\theta}$ can best be explained by the use of Fig. 1-B where q, the infiltration air, is assumed to enter the top of the door space, and as more air infiltrates, this boundary surface or interface proceeds downward at a velocity ν. The change in volume of the air entering or leaving the door segment with respect to time is ν As which is $\frac{dq}{d\theta}$.

The solutions of equations B-10 and B-11 were obtained by trial and error and the results are given in Fig. 2B.

We wish to thank Prof. Min for his planning in the early stages of this study which greatly facilitated the experimental program.

1761

No. 1761

Evaluation Procedure for Odor-Control Methods

WILLIAM F. KERKA
Member ASHRAE

Control of odors within air-conditioned spaces has been more of an art than a science. Modern engineering practice demands finer and finer control, and the effective and economical removal of odors is becoming an increasingly important factor in enclosed environments. This is especially so where the use of fresh, outdoor make-up air would be reduced to a minimum, and the recirculated indoor air picks up odors that are generated in the occupied spaces.

As part of the environmental program at the ASHRAE Research Laboratory, the study of air-conditioning odors has achieved an important position—one which should give much-needed information to the design engineer. It is the pur pose of this paper to examine a procedure which can be used to evaluate odor-control methods such as odor modification, sorption and oxidation. The term modification will be used to imply either masking, counteraction or cancellation.

Possible methods of evaluation — There are numerous methods of odor evaluation and each is based on the use of the human olfactory system. A few of the possible methods are outlined as follows:

1. **Use of a rating scale with numbers and word description related to the intensity of an odor.**
2. **The dilution technique, whereby a sample of odorous air is diluted with odor-free air until a threshold level is reached.**
3. **The comparison of an odor level with a series of samples of varying concentration, and matching the unknown against one of the samples.**

The above can be the result of either initial, immediate impressions or evaluations made after

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William F. Kerka is a Research Engineer at ASHRAE Laboratory. This paper has been prepared for presentation at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961.

long periods of exposure to the odorant.

TYPES OF ENVIRONMENTS

Studies made under actual field conditions; for example, in the occupied space of a building. Non-simulated, in-space, dynamic system.

Use of test rooms where the environmental conditions (temperature, humidity and odor concentration) can be controlled carefully. Simulated, in-space, static or dynamic system.

Odorous environment confined to an air stream with evaluations made at sniff ports. A dynamic, out-of-space, simulated system.

Confining the odorous air in containers, syringes, etc., for evaluation. A static, out-of-space, simulated system.

Many authorities in the field of odor research agree that the best way to evaluate odor intensity is to have the person (or at least the person's head) completely surrounded by the environment.[1]

The two 8 x 12 x 8-ft odor test rooms[2] at the ASHRAE Laboratory were found to meet this condition, and the complete air-conditioning system to each test space assured accurate control of the environments. In selecting a method of odor evaluation, it was concluded that a panel of from four to eight persons making a direct and immediate appraisal of the environment with a rating scale could obtain the most data in the least amount of time.

Odor-control methods and odor-

[1] Exponents refer to References.

Fig. 1 Mechanical smoker and recirculating system in test room

ants selected for study – A survey of manufacturers of odor-control equipment was made at the outset of the program to ascertain the availability of products on the market. The products used for study are listed in the panel below.

Method of Odor Control	Product	Description
1. Modification	a. Product "A"	Non-reactant, volatile liquid
	b. Product "B"	Non-reactant, volatile liquid
2. Sorption	a. Activated coconut-shell carbon	Granular #6x 10 mesh, 50-min accelerated service life by U. S. Government chloropicrin test
3. Oxidation	a. Ozone from small ozone - generating lamps	O_3 Gas

The following odorants were selected in the order of their importance, tobacco smoke usually being the most common offender in odor problems. It was also desirable to choose three different types of odorants – natural, synthetic multi-component and synthetic single-component.

Odorants	Description
1. Fresh Cigarette Smoke	Generated by mechanical puffer. A natural odor.
2. Synthetic Kitchen Malodor[3]	Composed of a mixture of pure compounds and essential oils. A synthetic, multi-component odor.
3. Iso-Valeric Acid	A pure vapor which resembles the smell of body odor. A synthetic, single-component odor.

Odor introduction apparatus – Apparatus used for producing cigarette smoke is shown in Fig. 1. One unit was located in each of the two test rooms. The normal puffing time for a king-size, filter-tip cigarette was about 12 min, although only the first ten min of this time were used for odor introduction. The mean out-of-the-pack weight of six randomly selected cigarettes was 1.155 gm, and the mean weight loss for the ten-min burning period was 0.782 gm. Most of the visible smoke generated came from the free-burning end.

Synthetic kitchen malodor was introduced into the test rooms by evaporation from a wick dipped in a bottle of the fluid. For this purpose the full-strength odorant was diluted 50% with distilled water. The wicks and bottles were placed in the small recirculating system located in each odor room as shown in Fig. 1. In an early series of tests, the malodor was introduced in the test space by spraying from an aerosol can.

Iso-valeric acid vapor was in-

troduced into the air supply duct to each test room by bubbling nitrogen through two gas-washing bottles coupled in series and containing the liquid acid.[3]

Operation of odor-control apparatus – Odor modifiers were introduced into the test rooms by evaporation from wicks in bottles located in the recirculating system. Because of possible fractionation during the evaporation process, the liquids (as well as the kitchen malodor) were changed for fresh batches about every fifth test.

Activated carbon in the sorption studies was used in canisters mounted on the inlet side of the fan of the small recirculating system that is shown in Fig. 1. An empty canister was used in one of the rooms so that the appearance of each room was the same to the panel member.

A mercury-vapor, ozone-generating lamp also was mounted in the recirculating system in each test room. The lamp (G.E. No. 0Z4S11) was operated in accordance with the manufacturer's specifications.

TEST PROCEDURE

The procedure used in this study was basically one of introducing a measured amount of malodor into each of the two test rooms. In one of the test rooms, the odor-control agent was allowed to act on the malodor for a given period of time, after which the odorous environments in each test space were evaluated by a group of subjects. In some preliminary studies, a rating scale that had been used in the past[2] was utilized for the evaluation of the odor levels as follows:

Level	Word Description
0	No odor
1	Threshold, recognition
2	Definite
3	Strong
4	Overpowering

However, since the odor levels normally encountered in air-conditioned spaces are seldom above the 2 or 3 level, the levels used in the study were kept within the lower portion of the scale. This, then, restricted the number of scores (even though half scores were permitted; for example, a 1½ or 2½) from which to evaluate. In order to broaden the choices for noticeable differences in level, a 0-to-10 point rating scale was adopted, with no associated word description other than the fact that 0 meant no odor, 1 was threshold (recognition threshold), and 10 was considered a maximum level.

A description of a developed test procedure is noted in the following steps (using cigarette smoke as the malodor and an odor modifier as the odor-control agent):

1. The test rooms are thoroughly purged with odor-free, conditioned air until the temperature and humidity in each test space is maintained at 75 DB and 50% relative humidity, respectively, (all tests were conducted at these conditions).

2. The external air supply then is shut off and a cigarette is placed in the smoking apparatus in each test room for exactly ten min. At

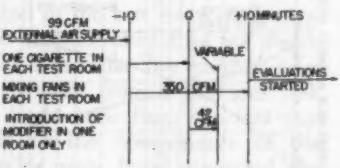
the beginning of the smoking period, the mixing fan (propellor fan not shown in Fig. 1) in each test space is started in order to assure complete mixing of malodor and control agent and to prevent stratification.

3. At the end of the smoking period, the cigarettes are removed and the bottle of odor modifier unsealed and placed in the small recirculating system of one of the rooms, and the recirculating fan turned on for a pre-determined period of time (this time determines the ratio of modifier concentration to malodor concentration). The mixing fans still operate during this period.

4. The bottle with odor modifier then is removed and the recirculating system shut off. The mixing fans continue to operate until ten min after the odor introduction was stopped.

5. After this period, the mixing fans are shut off and the subjects are called to evaluate the odor levels in each test room.

A schematic drawing of this static, batch-type test procedure is shown as follows, assuming that 0 time is the end of the odor-introduction period.



Odor evaluation procedure – Usually four runs were made during the day. The operator of the test rooms would select, at random, one or the other rooms to be treated with the odor-control agent; which room was being treated was unknown to the panel members. After the odor introduction and treatment cycles in the rooms were completed, the panel members were called. Half of the group entered one room first (one at a time) and the other half entered the other room first. The subject walked around the room at least once, sniffing several times, and then left. Upon leaving the first room (or in some cases while still in the room) he recorded on his score sheet the following odor-level impressions:

1. Over-all level – Lumped impression of items 2 and 3.
2. Malodor level – Cigarette, kitchen, or iso-valeric acid.
3. Background level–Odor modifier, activated carbon, or ozone.

After waiting for at least one minute, the panel member entered the other test room and went through the same procedure. The panel member was always told before a test what malodor and control agent were being used, but not, of course, which room was treated. All evaluations were made within a ten-min period or less.

The results of each test were tabulated and replicate tests were analyzed statistically (by the analysis of variance)[4] to determine if any significant reduction in the malodor level occurred when treated by the control agent. A confidence

level of 95% was considered significant.

The term "background level" was simply a column heading under which the panel members scored the odor level of the modifier (product A or B) or ozone, depending on what was used for a test. During the sorption studies the panel members were instructed to seek out any odor other than the malodor and score it as the background level.

The rooms themselves (if closed for long periods of time) have a characteristic woody odor slightly above threshold level, but this was not detectable during a test run when a malodor level of from 3 to 5 (on the 0-to-10 point scale) was used, or even when it was lowered by the control agent to a level of 1 or 2. However, daily airing of the rooms and standing over the week-end with doors opened reduced this problem to a minimum. All the surfaces of the rooms were washed with mild detergent and warm water periodically.

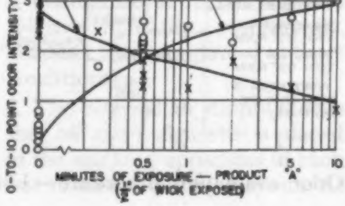
PRESENTATION OF RESULTS

Application of test procedure using odor-modifier agents – The results of the studies on the Product "A" odor modifier and cigarette smoke are shown in Fig. 2 (a logarithmic scale was used only for convenience). To ascertain at what proportions of malodor and modifier concentration studies should be conducted for the most effective control, the concentration of the product was varied by the length of time the wick was exposed to the air stream (from 15 sec to ten min), while the concentration of cigarette smoke remained constant (one cigarette burned in each of the two test rooms). The arithmetic means of the evaluations of malodor level and Product "A" level (background) made by the panel members then were plotted to obtain two distinct curves. The curves were drawn to fit the points by "eye." The values at 0 exposure time represent the levels in the test room with cigarette malodor only, and show the range of scores from test to test when the physical concentration remained constant. At the point where the two curves crossed, additional tests were made to determine if the reduction in malodor level was statistically significant compared to the room with malodor only. The results of five runs are shown as follows:

Malodor – Cigarette smoke – one burned in each test room for ten min.

Odor-Control Agent – Product "A," 0.5-min exposure, weight

Fig. 2 Odor level of product "A" and cigarette smoke versus exposure time of product "A"



loss = 0.0368 to 0.0507 gm (minimum and maximum for five runs), wick ½ in. above top of bottle.

the panel member was never aware of which room was being treated. He was told also that in some special tests both rooms might or might not contain the control agent.

Run Number	Cigarette Malodor Only: Over-all Level	Malodor Level	Background	Cigarette Malodor plus Product "A": Over-all Level	Malodor Level	Background
CW-9M	2.9	2.3	0.8	3.6	1.9	1.9
CW-13M	3.2	3.0	0.4	3.2	1.8	2.3
CW-14M	3.3	2.6	0.7	3.3	1.8	2.2
CW-15M	2.9	2.6	0.3	3.1	1.3	2.6
CW-16M	2.9	2.5	0.5	3.2	1.5	2.1

Each value is the mean of five individual scores.

For the five replicate tests, the reduction in malodor level is significant by the 99% confidence level. However, there is virtually no difference in over-all level between the treated room and the room with malodor alone. To operate to the right of the crossover point (Fig. 2) further would reduce the malodor level, but only at the expense of increasing the odor level of the product above that shown in the tabulation. This would result in masking (superimposing one odor for another) rather than the counter-action (reduction in level of both malodor and odor modifier) that is desired.

The fact that some background odor level of the product was recorded in the test room with cigarette malodor only does not necessarily mean that any residual odor was present from a previous test, but could arise from the fact that some of the components of the malodor may have reminded the panel member of the product odor. It should again be pointed out that

The same procedure of determining the cross-over point of operation for Product "A" and kitchen malodor was used. The results of five runs at slightly to the left of this point showed a significant reduction in malodor level. However, the odor of the product in the treated room was again evident. To assure that the kitchen malodor was not over-treated; that is, too much odor modifier used, a series of five runs was made with a lower concentration of Product "A" (to the left of the cross-over point). The analysis indicated that the reduction in malodor level was not statistically significant. On the other hand, the odor level of the product (background) in the treated room was also lower than in the preceding tests.

Application of procedure to Product "B" odor modifier—The results of the studies on Product "B" odor modifier and cigarette smoke are shown in Fig. 3. It was found that

the physical concentration of this product for recognition threshold was considerably less than that of Product "A." The amount of Product "B" evaporated from the ½-in. exposed wick was so great that a short enough exposure time was not possible. As a result, the wick was submerged and evaporation from the surface of the liquid was utilized. Because for a given exposure time the weight loss of Product "B" varied considerably from test to test, the results are plotted on a weight-loss basis rather than on an evaporation-time basis. The six tests that were run close to the crossover point were tabulated and analyzed: Malodor – Cigarette smoke – one cigarette burned in each test room for ten min. Odor-Control Agent – Product "B," weight loss 0.0030 to 0.0061 gm, no wick exposed.

In this study the panel members were instructed that Product "B" would be scored as the background level. Also, as in the previous runs, evaluations were begun upon the termination of the ten-min mixing period. Analysis of the data indicates that the probability of the results of the six tests occurring by chance is less than 0.001; hence, the reduction in the malodor intensity by the modifier is significant by the 99.9% confidence level. The relatively same over-all level for the treated and untreated rooms again is evident. The product level (background) in the treated room also is noticeable. The rapid rise in the product level at the crossover point would, in actual practice, make proper control difficult; however, the exact shape of this curve needs further study.

The results of the studies on Product "B" and kitchen malodor near the cross-over point again indicated a significant reduction in malodor level, but with the noticeable odor level of the product.

Run Number	Cigarette Malodor Only: Over-all Level	Cigarette Malodor Only: Malodor Level	Cigarette Malodor Only: Background	Cigarette Malodor plus Product "B": Over-all Level	Cigarette Malodor plus Product "B": Malodor Level	Cigarette Malodor plus Product "B": Background
CB-2M	2.6	2.5	0.2	3.4	1.4	2.9
CB-3M	2.5	2.3	0.4	3.1	1.2	3.5
CB-4M	3.4	3.3	0.2	3.7	1.7	3.0
CB-5M	3.3	3.0	0.2	3.1	1.9	2.2
CB-9M	3.7	3.5	0.1	3.3	1.8	2.5
CB-10M	3.3	3.0	0.6	2.4	2.0	0.9

Each value is mean of five individual scores.

Application of test procedure to activated carbon – The fact that activated carbon will adsorb odorous vapors has long been established.[5] In its application, there are a number of problems that must be considered[6]:

1. Operating life – dependent, among other things, upon its rate

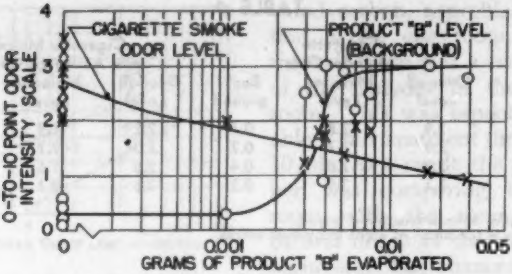


Fig. 3 Odor level of product "B" and cigarette smoke versus weight loss of product "B"

of loading, its exposure time and the type of vapor involved.

2. Amount of treatment – the weight of activated carbon, and the amount of air and vapor treated by it to produce significant odor reduction.

3. The economics of its use – maintenance, reactivation, replacement, etc.

Items 1 and 3 are beyond the scope of this paper since they are related more to conditions in the field than to laboratory experimentation. The test procedure will, therefore, be applied to item 2. The activated carbon was used in canister form (5 lb in a canister). The canisters were mounted on the inlet side of the fan of the recirculating system in each room as previously described. In these studies all test runs were again static, batch-type processes. No additional mixing (other than that produced by the recirculating system) was utilized in the test rooms since these tests preceded the time when it was concluded that additional mixing should be used. In a sense, however, this subjected the operation of the carbon canister to the conditions most likely encountered in actual use.

A series of four tests all run at the same treatment time was analyzed. In the case of the background level, the panel members were instructed to rate any odor other than the malodor as background. (See Table A.)

The statistical analysis of the four replicate tests indicates that the reduction in malodor level by the carbon treatment was not significant. However, it was noted that one of the panel subjects in this test series was consistantly scoring opposite to the other seven members. Elimination of this person's score gave test results that showed a significant reduction in cigarette malodor by the 95% confidence level. The relatively low background level for each condition is evident in the preceding scores.

To gain a better perspective of the treatment time necessary for

TABLE A

Run Number	Cigarette Malodor Only: Over-all Level	Cigarette Malodor Only: Malodor Level	Cigarette Malodor Only: Back-ground	Cigarette Malodor plus Activated Carbon: Over-all Level	Cigarette Malodor plus Activated Carbon: Malodor Level	Cigarette Malodor plus Activated Carbon: Back-ground
CC-1	2.9	2.8	0.1	2.2	1.9	0.8
CC-2	2.9	2.7	0.2	3.0	2.7	0.7
CC-3	2.8	2.6	0.4	2.5	1.6	1.2
CC-4	3.1	2.9	0.3	3.6	3.1	0.9

All runs have 20-min treatment period.
Each value is the mean of eight individual scores.

TABLE B

Run Number	Treatment Time Min	Cigar Malodor Only: Over-all Level	Cigar Malodor Only: Malodor Level	Cigar Malodor Only: Back-ground	Cigar Malodor plus Activated Carbon: Over-all Level	Cigar Malodor plus Activated Carbon: Malodor Level	Cigar Malodor plus Activated Carbon: Back-ground	Significance of Malodor Reduction 95% Confidence Level
MC-1	12.5	3.9	3.8	0.3	3.1	2.9	0.3	No
MC-2	17.5	3.8	3.6	0.4	3.2	2.9	0.5	Yes
MC-3	22.5	4.2	4.1	0.5	2.5	2.1	1.2	Yes
MC-4	27.5	3.4	3.2	0.7	2.6	1.9	1.1	Yes
MC-5	32.5	3.3	3.1	0.3	1.7	1.0	1.1	Yes

Each value is the mean of six to eight individual scores.

TABLE C

Run Number	Treatment Time Min	Iso-Valeric Acid Vapor Only: Over-all Level	Iso-Valeric Acid Vapor Only: Malodor Level	Iso-Valeric Acid Vapor Only: Back-ground	Iso-Valeric Acid plus Activated Carbon: Over-all Level	Iso-Valeric Acid plus Activated Carbon: Malodor Level	Iso-Valeric Acid plus Activated Carbon: Back-ground	Significance of Malodor Reduction 95% Confidence Level
VC-1	15	2.7	2.2	0.5	1.8	0.6	1.3	Yes
VC-2	15	3.2	2.4	1.1	2.5	2.1	0.6	No
VC-3	15	3.2	2.8	0.4	2.2	1.4	0.8	No
VC-4	20	2.5	2.1	0.7	1.3	0.4	1.0	Yes
VC-5	20	2.6	2.2	0.6	1.8	0.9	1.2	Yes
VC-6	25	2.4	1.8	0.8	1.9	0.8	1.3	Yes
VC-7	25	2.7	2.3	0.7	1.4	0.9	0.8	Yes

Each value is the mean of eight individual scores.

odor reduction, studies with cigar smoke odor were made with varying lengths of treatment time. In these tests, the cigar was burned for only 2.5 min in each test room, since it was found that this produced a high enough odor level for evaluation purposes. The next tabulation shows the results of five tests with the treatment time listed in each case. (See Table B.)

Since each test has a different treatment time, an analysis of the individual tests was made instead of an aggregate analysis. The significant reduction in malodor level is noted at the 17.5-min treatment period and above. The progressively lower cigar malodor level and over-all level with increased

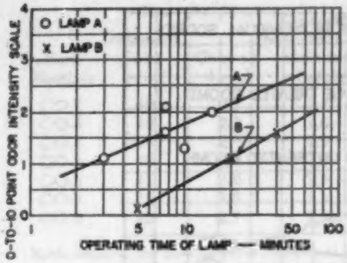


Fig. 4 Odor intensity level of ozone versus operating time of ozone-generating lamp

treatment time also are evident in the room using activated carbon.

The same procedure (a different carbon canister was used for each malodor) was used in applying the test procedure to activated carbon and iso-valeric acid vapor. The supply air was introduced into each test room at a rate of 99 cfm with a malodor concentration of 0.0000218 oz/1000 cu ft (0.0053 ppm by volume) for a period of five min for each run, and the recirculation through the carbon canisters was varied from 15 to 25 min. The tabulation of seven runs is presented in Table C.

It would appear that a treatment time of 15 min was not adequate in significantly reducing the odor level of the iso-valeric acid vapor. For the 20-min period of adsorption and above, however, significant reduction in both overall and malodor level was noted for each test run. The low background level is again evident.

In the studies treating synthetic kitchen malodor with activated carbon, a problem arose, in that, while one room was being treated, there was a natural decay of the malodor in the untreated room. This was especially noticeable if the treatment time exceeded 10 min. As a result, the panel member was comparing the treated room with the room in which natural decay of the malodor was occurring. The chances for recognizing differences would thus be lessened. There was little evidence of this rapid natural decay with the tobacco odor and the iso-valeric acid vapor. The analysis of the aggregate of six tests (treatment time, 10 to 20 min) showed the reduction in malodor level to be significant by the 99.9% level.

Application of procedure to ozone — At the beginning of the ozone studies, an evaluation of the intensity level of the generated ozone versus the operating time of the lamp (original lamps referred to as lamps A) was made. This is shown as line A in Fig. 4. Unfortunately, during this period no means of measuring the ozone concentration was available. In later studies, two other lamps (henceforth referred to as lamps B) were used (one for each room) when the original two were found by sensory appraisal to be producing a lower level for a given operating time. The physical concentration of ozone produced by each lamp B was determined by noting the decay of a special rubber strip,[7] and also by using the oxidation-reduction principle of a potassium iodide solution.[8] The degree of decay of the rubber

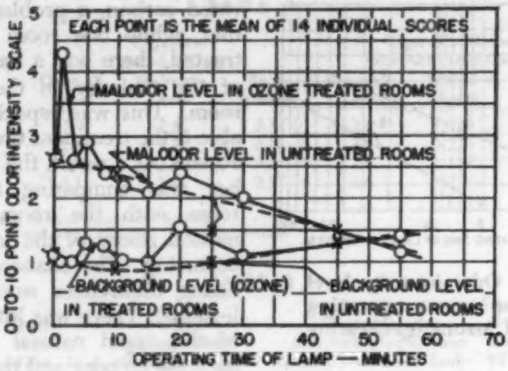


Fig. 5 Malodor level (iso-valeric acid) and ozone level in treated and untreated room versus time of ozone treatment

strip for a given time of exposure was indicative of the ozone concentration in ppm by volume. It was found that continuous operation of a lamp B in the sealed-off static test room produced an equilibrium concentration of ozone of about 0.002 to 0.003 ppm (2 to 3 ppb). This equilibrium concentration is reached after about one hour operating time of the lamp. Half this concentration is reached in approximately ten min.[9] From Fig. 4, line B, the recognition threshold level for ozone was obtained after 20 min operating time, or a physical concentration of about 0.0015 ppm. This is less than that found by other investigators.[10,11]

Virtually the same test procedure was used in the ozone studies as in the preceding studies; that is, the introduction of the malodor in each test room, and the treatment of the environment by the control agent in one of the rooms for a given period of time. It should be mentioned again that throughout the test program special runs were made unknown to the panel members, whereby either or both rooms were treated simultaneously, or both were not treated. This served to check the responsiveness of the subjects to make sure that they were not partial to either room.

The panel members were familiarized thoroughly with the odor quality of ozone generated by the lamps, and were instructed that this odor characteristic would be scored as the background in the three-component analysis method used. Seven tests using ozone from lamp A as the control agent and cigarette smoke as the malodor were run with a varying treatment time.

An analysis of the four replicate tests for 15 min of treatment

Run Number	Treatment Time Min	Cigarette Malodor Only: Over-all Level	Cigarette Malodor Only: Malodor Level	Cigarette Malodor Only: Background	Cigarette Malodor plus Ozone: Over-all Level	Cigarette Malodor plus Ozone: Malodor Level	Cigarette Malodor plus Ozone: Background
CO-2	10	2.3	2.2	0.8	2.6	1.9	1.0
CO-4	10	2.3	1.6	1.0	3.0	1.8	1.4
CO-1	15	2.1	1.9	0.7	2.0	1.0	1.8
CO-3	15	3.4	3.3	0.3	3.0	1.5	1.7
CO-6	15	3.0	2.4	0.6	2.8	2.2	1.1
CO-7	15	3.5	3.3	0.3	3.0	2.0	1.5
CO-8	20	2.9	2.6	0.4	2.8	1.4	1.8

Each value is the mean of five individual scores.

time shows a significant reduction in cigarette malodor level. The ozone in the treated room was perceptible as shown in the last column.

The studies using iso-valeric acid vapor were conducted over a longer period of operation time (up to 55 min) of the ozone generating lamp (Lamp A) to assure an adequate reaction time between the ozone and the malodor. Instead of treating one room and comparing it with the untreated room, both rooms were treated with ozone for the first group of runs, and then both rooms were untreated for the second group of runs. The comparison, therefore, was made on this basis. The fact was not known to the panel members.

The test results for the treatment of the malodor with ozone are shown in Fig. 5. Here a direct comparison of the plotted levels was made between the rooms with iso-valeric acid vapor alone (untreated) and then with the vapor treated by the ozone. The fact that the reduction in malodor intensity due to ozone treatment is no greater than the natural decay (with time) of the malodor in the untreated rooms is evident. The slight background level of ozone reported in the untreated rooms again could arise from the fact that some of the characteristic odor notes of the malodor reminded the panel members of the ozone. The over-all intensities (not shown in Fig. 5) for both the treated and untreated rooms were about the same for each condition.

In the studies using synthetic kitchen malodor and ozone, the test procedure was identical to the one just described for iso-valeric acid except that the malodor was introduced into each room by spraying from an aerosol can for a two-sec period. The test results are plotted in Fig. 6. The abscissa in Figs. 5 and 6 (operating time for ozone lamp A) applies to the treated rooms only, the untreated rooms (with malodor alone) being allowed to stand for the same length of time before evaluations were begun.

From Fig. 6, there appears to be no significant reduction of the malodor level by the ozone as compared to the natural decay of the malodor alone, except possibly at the longest treatment time. However, here the level of ozone is so

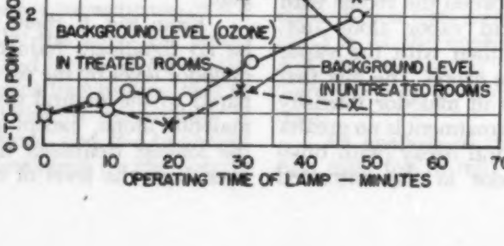
high that masking of the malodor probably takes place. The steady rise in ozone level is apparent with increased operation time of the lamp.

Test runs using ozone lamps B — When ozone lamps B were installed, a series of repeat runs was made using cigarette smoke and ozone, the latter at a concentration of about 0.002 to 0.003 ppm. The mixing fans also were used in the test rooms to comply with the test procedure recommended at the beginning of this paper. The results of four identical tests showed that there was no significant reduction of the cigarette malodor when treated with ozone as compared to the cigarette malodor alone.

Although all procedures of testing described so far are based on immediate impressions, it was concluded that possible exposure to ozone by the panel members over a period of time might cause some fatiguing of the olfactory system and hence a lower sensitivity to the malodor. To study this effect, cigarette smoke odor was generated in one of the static test rooms. The panel members entered this room (all at one time) and made an immediate evaluation of the odor level. The panel then entered the other room where they remained for exactly ten min.

In one series of tests the second room had been purged thoroughly with fresh air; and in the second series, it contained a concentration of ozone of about 0.002 to 0.003 ppm. This fact, of course, was unknown to the panel. After the ten-min exposure, the subjects re-entered the first room and again evaluated the cigarette smoke odor level. From previous studies it had been found that the odor level of

Fig. 6 Synthetic kitchen malodor level and ozone level in treated and untreated room versus time of ozone treatment



cigarette smoke did not decay appreciably during the ten-min period following odor introduction. The following tabulation shows the results of this test:

Ten-min exposure in room with ozone at concentration of 0.002 to 0.003 ppm by vol	
Initial Odor Level	Final Odor Level
3.9*	2.8
4.0	3.0
3.6	3.0
4.0	3.3
Mean of 4 Tests 3.9	3.0

* Each value is mean of four subjects.

It has been noted in the past that there is a tendency for the second judgment of a given level to be lower than the first (for this reason half the subjects normally entered one room and half the other room first). This point is evident in the preceding results, but the fact that, after exposure of an ozone concentration 0.002 to 0.003 ppm, the second judgment was no lower than the one made after exposure to odor-free air, indicates that the exposure time or concentration of ozone was not great enough to cause any fatiguing of the olfactory receptors.

To further study the exposure to ozone for an extended period, the panel members remained in the test room for a ten-min period. In one case the room contained only cigarette malodor (one cigarette puffed for ten min) and in the other case ozone again at a concentration of 0.002 to 0.003 ppm was present with the malodor. Again this was unknown to the panel. The results of this study are shown in Fig. 7. Each point is the mean score of four subjects partaking in three identical tests for each condition and represents the

Ten-min exposure in Odor-Free Room	
Initial Odor Level	Final Odor Level
3.5	3.1
4.4	2.8
3.8	3.1
Mean of 3 Tests 3.9	3.0

initial cigarette malodor level and evaluations of malodor level made at one-min intervals thereafter. The effect of adaptation is evident[2] in both cases. The adaptation rate for each of the two conditions is nearly the same until about seven min, where there is a hint of a continued decline when exposed to the malodor plus ozone. It would appear that exposure to this concentration of ozone and malodor must be greater than a ten-min period before fatigue to the olfactory receptors exceeds the adaptation that occurs when exposed to the malodor alone.

DISCUSSION

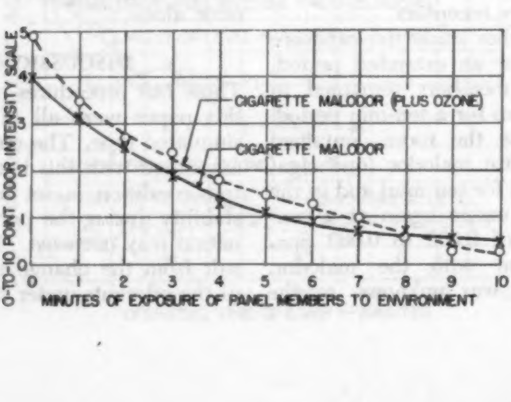
Those test procedures presented in this paper were all of the static, simulated type. The main problem associated with this is that, once a test condition is set up, absolute stability during the period of evaluation may not exist. This can result from the change in character of the odorants under study due to

adsorption and desorption on the room surfaces, progressive dilution of the test room environment when the subject enters through the door from the outside, and the natural decay with time that complex odorants may exhibit. Having all the subjects enter a test room at one time would be advisable. In this study, however, the subjects (staff members of the Laboratory) were called from their normal work routine and in some cases were not able to respond immediately. Therefore, a maximum evaluation period to ten min was established during which all subjects completed the evaluation of each room, one at a time. To minimize any unstable effects, it would be necessary to establish a strict schedule of evaluation in the shortest possible time.

The advantages of the static system are that it is relatively easy to set up, and the amounts of odorant and odor-control agent introduced into the system can be measured readily. In the dynamic system, continuous control of all variables must be exercised to maintain an equilibrium condition. The described test setup can be used easily without physical alteration as a dynamic system except for its application to the study of odor control by sorption. In this case the adsorbent would have to be relocated in the air supply system and the odorant introduced upstream from this point. For the other cases, however, conditioned air is supplied continuously to the room, and controlled rates of modifier or ozone are fed into the test space through the small recirculating system located in the space. The odorant also is fed into the room at a continuous, measured rate.

Odor introduction techniques – The method of introducing the cigarette smoke odor appeared reproducible. Of the six randomly selected cigarettes for weighing, the measured loss for the ten-min burning period ranged from 0.6436

Fig. 7 Adaptation to cigarette malodor and malodor plus ozone



to 0.8739 gm. Each of these extremes deviates less than 33% from the 0.7820-gm mean, or less than a just-noticeable-difference on an odor-perception basis. Since most of the smoke from a humanly puffed cigarette comes from the free-burning end, the mechanical puffing action complied with this.

Concentration of the iso-valeric acid vapor could be determined accurately, and past studies[2] have shown agreement within 10% of measured and calculated values.

Introduction of the synthetic kitchen malodor by evaporation from a wick dipped in the diluted liquid proved unreliable because of fractionation. It is recommended that the liquid be aerosolized into the space for absolute reproducibility. Spraying the malodor from an aerosol can for some of the tests appeared to give a more consistent quality and level; but in addition, a means of measuring the exact weight loss should have been utilized.

Introduction and measurement of the odor-control agents – The primary purpose for introducing the odor modifiers by evaporation from a wick was to comply as closely as possible to their actual use in an air-conditioning system. It can certainly be argued that more rigid control should have been exercised. However, the method used did bring out a notable point; namely, for a given exposure time of Product "A" the weight loss varied only ±21% (less than a just-noticeable difference).

One of the problems in measuring the concentration of ozone with the special rubber strip is that it must be exposed for a sufficient period of time to obtain a measurable amount of decay. As a result, the reading obtained is not an instantaneous value, but rather the integrated mean value over the period of exposure. The instrument using the oxidation-reduction of potassium iodide[8] is readily responsive to changes in concentration, and hence gives near-instantaneous values. Witheridge and Yaglou[11] have shown that the absolute humidity affects the disappearance of ozone in a room, but since all tests in this study were made at the same condition (75DB and 50% RH) ±2.5F, the humidity effect was negligible. Also with the low concentrations used in this study, the degeneration that occurred during the five to ten-min evaluation period after ozone introduction was stopped appears to be of small magnitude.[9]

The term "oxidation" has been used in conjunction with ozone as a possible means of odor reduction. The results of the tests conducted, especially Figs. 5 and 6, would indicate that masking probably occurs if enough ozone is generated. Further support of this fact is given in reference 11. The maximum allowable concentration of ozone recommended for long-term exposure is given as 0.1 ppm.[12] At this concentration (measurements were made near the ozone generating lamp in the recirculating system) ozone has a strong, pungent odor. Even at the measured concentrating of 0.003 ppm, the characteristic

odor is quite evident and it would seem unreasonable to use much higher levels in air-conditioning systems.

Odor evaluation – In applying the test procedure, one basic fact was determined: does a reduction in malodor level occur when treated by the control agent? Although the study of odor character notes by profile analysis for blends of odorants would more completely describe the problem,[18] it cannot be denied that, if one criterion is to be chosen, the analysis of odor level stands out as one of the foremost. A study of the quality of the environment before and after treatment is of equal importance, but since this type of evaluation involves preference ratings, it is beyond the scope of this paper. It is planned that the examination of profile analysis and preference rating be incorporated in a future, continued study of odor control methods.

In using the three components to evaluate the levels in the odor test rooms; that is over-all, malodor and background, the over-all level was not necessarily the arithmetic sum of the other two components. This can be resolved more clearly by considering two identical concentrations of the same odorant. For instance, if each were at a sensory level of 3, combining them (or doubling the concentration) would not give an over-all level of 6. The panel members were instructed thoroughly on this point. The use of the term "background" was perhaps not the best choice. Since one of the test rooms did not contain the control agent and one did (but in some special tests both rooms did not or both did), the choice of one term to apply to all conditions was not an easy one.

However, from continuous discussions with the panel members, there was no doubt in their minds about scoring the control agent as "background" when the agent itself had a characteristic odor. In the sorption studies the term was less definable. If the fact that some background product level was reported in the test room with only malodor resulted from the product odor being carried on the subject's or operator's clothes from the treated to untreated room, then the procedure of treating one room and not the other should be altered so that both either are treated or untreated. Comparisons would then have to be made from one test to the odor of the modifier also is same test.

CONCLUSIONS

1. Procedure of testing described can be applied to a number of odor-control methods.
2. Main deficiency of the static type of test used is that absolute stability of the conditions may not exist during the treatment and evaluation periods.
3. Significant reduction in malodor level can be achieved by treatment with odor modifiers, but the odor of the modifier also is perceptible.
4. Significant reduction in malodor level can result from treat-

ment with activated carbon.

5. Reduction of odor levels by ozone at concentrations slightly above threshold is not statistically conclusive by the results presented in this study.

6. Described procedures are to be considered exploratory in nature, and should serve as a guide for further experimentation in the broad field of odor control.

REFERENCES

1. Past discussions with Dr. Bernice Wenzel, University of California Medical School.
2. "Temperature and Humidity Effect on Odor Perception" by W. F. Kerka and C. M. Humphreys, HP & AC Journal Section, April, 1956, p. 129.
3. Synthetic Kitchen Malodor per specifications of General Services Administration, Federal Supply Service, in Interim Federal Spec. No. P-D-00200 (GSA-FSS) dated September 27, 1956, by C. J. D'Angio.
4. An Introduction to Statistics in the Medical Sciences, by H. C. Batson, Phd., Burgess Publishing Co., Minneapolis, 3rd Printing—1958, p. 22.
5. ASHRAE GUIDE AND DATA BOOK, 1960, Chapter 7, pp. 82-87.
6. Air Pollution Abatement Manual—Odor Measurement and Control. Mfg. Chemists Assoc., Inc., Chapter 13, page 11, by Amos Turk, Ph.D.
7. B. F. Goodrich Co., ozonometer rubber strips.
8. Most Development Co., ozone meter and recorder.
9. Communication with James R. Luck, Senior Chemist, Minneapolis-Honeywell Regulator Co.
10. Communication by Dr. Herbert Stockinger, O.H.F.H., U.S. Public Health Service.
11. "Ozone in Ventilation—Its Possibilities and Limitations" by W. N. Witheridge and C. P. Yaglou, ASHVE Transactions, 1939, Vol. 45, p. 509.
12. American Conference of Governmental Industrial Hygienists, "Threshold Limit Values for 1958", A.M.A. Archives of Industrial Health, August, 1958, pp. 178-182.
13. "Methodology of the Flavor Profile", by L. B. Sjostrom, S. E. Cairncross and J. F. Caul, Food Technology, 1957, Vol. XI No. 9 pp. 20-25.

DISCUSSION

E. SCOTT POINTER, Birmingham, Ala.: Would there be any significance in the test work if the panel members were smokers or non-smokers?

C. M. HUMPHREYS: Panel members who are smokers refrain from smoking for a twenty-minute period prior to the test. That length of time is sufficient to eliminate any effect of smoking.

RALPH T. ZINN, Denver, Colo.: Has an investigation been made of the use of negative ionization of the air as a method of odor removal?

C. M. HUMPHREYS: Some work was done at the Laboratory on the effect of ionization on comfort. We know of no work which has been done on the effect of ionization on odors.

FRED VOGT, Minneapolis, Minn.: From our experience along this line of removing odors and smoke, it was found that with the electrostatic filter the visible smoke color or the actual color tars could be removed. But it took the activated carbon filter to remove the so-called gases as part of the smoke, because if it were removed only with the electrostatic filter there remained a gas which left a sting in the air.

RALPH T. ZINN: What kind of odor control is used in atomic submarines? Only limited space is available for equipment on these units.

E. P. PALMATIER, Syracuse, N. Y.: I am almost certain that activated charcoal or activated carbon is used in these systems.

HARRY EBERT, Houston, Tex.: I would like some information on odor control in animal quarters where the predominant odor is ammonia.

JOHN BORRY, Minneapolis, Minn.: There are extensive animal quarters at one hospital; activated carbon was used there but it did not work on the ammonia.

1762

No. 1762

Daily Insolation on Surfaces Tilted toward the Equator

BENJAMIN Y. H. LIU

RICHARD C. JORDAN
Presidential Member
ASHRAE

Measurements of solar radiation have been made up to this time primarily on horizontal surfaces and data for sloped or vertical surfaces are extremely sparse. However, the solution of engineering and scientific problems involving solar radiation generally requires a knowledge of the solar radiation incident upon surfaces of various orientation. Therefore, a method is here developed which enables the solar radiation on sloped surfaces to be determined when the radiation incident upon a horizontal surface is known.

Although, with modification, the method proposed is probably applicable to surfaces of any orientation, the present research is restricted to surfaces tilted toward the equator (i.e. tilted toward the south for localities in the northern hemisphere and tilted toward the north for localities in the southern hemisphere) only. Solar energy collecting devices of current interest, with the exception of those equatorially mounted to follow the diurnal motion of the sun and those at outer space, are generally so orientated for maximum exposure to sunshine. Radiation of solar origin only is considered and thermal radiation, either from the ground or from the atmosphere, is not involved.

The subject of the present research has been investigated by various others[1,2,3] but the resulting empirical treatments of the prob-

Benjamin Y. H. Liu is Assistant Professor, Richard C. Jordan is Professor and Head, Dept of Mechanical Engineering, University of Minnesota. This paper was prepared for presentation at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961. It is the result of researches sponsored by grants from the American Society of Heating, Refrigerating and Air Conditioning Engineers and the National Science Foundation. Portions of the paper derive from the thesis of Assistant Professor Liu prepared in partial fulfillment of requirements for the Ph.D. degree.

[1] Jordan, R. C. and Threlkeld, J. L., "Solar Energy Availability for Heating in the United States," Heating, Piping and Air Conditioning, 25(12):111-121, December, 1953.

lem differ from the fundamental approach here followed.

THEORETICAL CONSIDERATIONS

Let H and H_t be respectively the daily summations of the total radiation incident upon a horizontal surface and a tilted surface, and let D be the daily summation of the diffuse radiation incident upon a horizontal surface. Since H_t includes not only the direct solar radiation from the sun and the scattered solar radiation from the various parts of the sky but also the solar radiation reflected onto the tilted surface from the ground, whereas H is composed of only the direct and diffuse radiation,

$$H_t = (H - D)\,R_D + D\,R_d + H\,R_\rho \quad (1)$$

where R_D, R_d and R_ρ are respectively the conversion factors for the direct, diffuse and ground reflected radiation and are defined by the following ratios:

$$R_D = \frac{\text{daily direct radiation incident upon the tilted surface}}{\text{daily direct radiation incident upon a horizontal surface, } (H - D)}$$

$$R_d = \frac{\text{daily diffuse sky radiation incident upon the tilted surface}}{\text{daily diffuse sky radiation incident upon a horizontal surface, D}}$$

$$R_\rho = \frac{\text{daily summation of the radiation reflected onto the tilted surface from the ground}}{\text{daily total radiation incident upon a horizontal surface, H}}$$

Thus the conversion factor, R, for the daily total radiation is given by the following equation,

$$R = H_t/H = (1 - \frac{D}{H})\,R_D + \frac{D}{H}\,R_d + R_\rho \quad (2)$$

Each of these conversion factors, R_D, R_d and R_ρ and the ratio $\frac{D}{H}$ will be separately considered.

1. Conversion factor for daily direct radiation – In order that an approximate analytical expression for R_D may be obtained, consider the following two special cases.

Special Case 1 – Surfaces Located Outside the Atmosphere of the Earth:

Outside the atmosphere of the earth, the solar radiation incident upon a horizontal surface is entirely direct radiation and therefore

$$R_D = H_{ot}/H_o \quad (3)$$

where H_{ot} and H_o are respectively the extraterrestrial daily insolation on the tilted and the horizontal surfaces. For this special case, R_D may be derived as follows.

The extraterrestrial solar radiation intensity, I_{oh}, on a horizontal surface at any instant is given by

$$I_{oh} = I_{on} \cos \theta_h \quad (4)$$

[1] The method of E. M. Brooks as described by Hottal, H. C., in "Performance of Flat-Plate Solar Energy Collectors," Space Heating with Solar Energy, Proceedings of a Course Symposium held at M.I.T., pp. 58-71, 1950.

[2] Mossman, J., "Solar Incidence Conversion Factor and Heat Storage Efficiency of Water," M.Sc. Thesis in Chemical Engineering, M.I.T., Cambridge, Massachusetts, September, 1950.

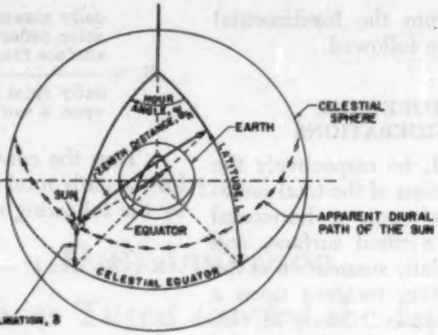


Fig. 1 Relationship between zenith distance, θ_h, latitude, L, solar declination, δ, and hour angle, ω

where

I_{on} = radiation at normal incidence at the outer limit of the atmosphere = solar constant, corrected for the variation of the distance between the earth and the sun from the mean distance, Btu/hr/sq ft

θ_h = incidence angle of the solar rays upon a horizontal surface = zenith distance = 90 deg − solar altitude angle

The incidence angle, θ_h, is a function of the latitude, L, the solar declination, δ, and the hour angle, ω, as shown in Fig. 1. The following basic astronomical equation is stated without proof,

$$\cos \theta_h = \cos L \cos \delta \cos \omega + \sin L \sin \delta \quad (5)$$

The extraterrestrial daily insolation, H_o, can be found by integrating Equation (4) with respect to time for all hours during which the sun is above the horizon, i.e. when $\cos \theta_h$ is positive. Since one hour is equivalent to 15 deg or $\frac{2\pi}{24}$ radians of hour angle, and since

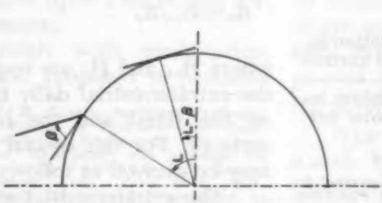


Fig. 2 Diagram showing that surface located at the latitude, L, tilted toward the equator at an angle β deg from the horizontal surface is parallel to a horizontal surface at the latitude L − β deg

I_{on} and δ are both approximately fixed quantities during one day,

$$H_o = I_{on} \int_{-\omega_s}^{\omega_s} (\cos L \cos \delta \cos \omega + \sin L \sin \delta)\, d\left(\frac{24}{2\pi}\omega\right)$$
$$= \frac{24}{\pi} I_{on} (\cos L \cos \delta \sin \omega_s + \omega_s \sin L \sin \delta) \quad (6)$$

where ω_s is the sunset hour angle in radians and is the hour angle at which $\cos \theta_h$ is zero. By Equation (5),

$$\cos \omega_s = -\tan L \tan \delta \quad (7)$$

It should be noticed that during the summer months and in the polar circle when the sun never sets, indicated by the fact that the product on the right hand side of Equation (7) has an absolute value greater than one, ω_s in Equation (6) should be replaced by π.

Similar to Equation (4) one may write for a tilted surface,

$$I_{ot} = I_{on} \cos \theta_t \quad (8)$$

where I_{ot} and θ_t are the extraterrestrial radiation intensity and the angle of incidence of the solar rays on the tilted surface. It can be seen from Fig. 2 that a surface located at a latitude, L, tilted β deg from the horizontal surface toward the equator is parallel to a horizontal surface at the latitude $L - \beta$. Therefore θ_t is the same as θ_h at the latitude $L - \beta$, or

$$\cos \theta_t = \cos (L - \beta) \cos \delta \cos \omega + \sin (L - \beta) \sin \delta \quad (9)$$

The extraterrestrial daily insolation, H_{ot}, on a surface tilted toward the equator can then be found similarly by integrating I_{ot} of Equation (8) with respect to time. Since the integration should be carried only over the period during which the sun is above the horizon and in front of the tilted surface, both $\cos \theta_h$ and $\cos \theta_t$ should remain positive within the limits of integration. If a sunset hour angle, ω_s', is defined by setting $\cos \theta_t$ to zero,

$$\cos \omega_s' = -\tan (L - \beta) \tan \delta \quad (10)$$

then the limits of integration should be either $(-\omega_s, \omega_s)$ or $(-\omega_s', \omega_s')$ whichever cover a smaller range of values. The following two equations for H_{ot} are obtained,

$$H_{ot} = \frac{24}{\pi} I_{on} [\cos (L - \beta) \cos \delta \sin \omega_s + \omega_s \sin (L - \beta) \sin \delta]$$
when
$$\omega_s \leqq \omega_s' \quad (11)$$

$$H_{ot} = \frac{24}{\pi} I_{on} [\cos (L - \beta) \cos \delta \sin \omega'_s + \omega'_s \sin (L - \beta) \sin \delta]$$
when
$$\omega_s' \leqq \omega_s \quad (12)$$

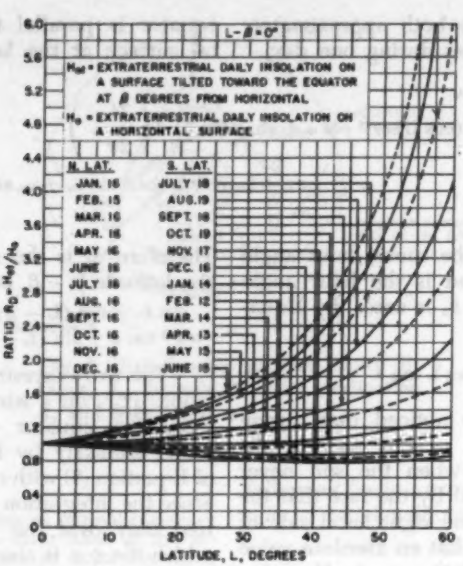


Fig. 3

Figs. 3 to 7 Ratio of the extraterrestrial daily insolation on a surface tilted toward the equator at an angle of β deg from the horizontal surface to that on a horizontal surface for $L - \beta = 0$ in Fig. 3, for $L - \beta = -10°$ in Fig. 4, for $L - \beta = -20°$ in Fig. 5, for $L - \beta = -30°$ in Fig. 6 and for $\beta = 90°$ (vertical surface) in Fig. 7

Corresponding to the above two equations, two expressions for R_D for the special case under consideration are obtained:

$$R_D = \frac{\cos(L - \beta)}{\cos L} \, \frac{\sin \omega_s - \omega_s \cos \omega_s'}{\sin \omega_s - \omega_s \cos \omega_s} \text{ when } \omega_s \leqq \omega_s' \tag{13}$$

$$R_D = \frac{\cos(L - \beta)}{\cos L} \, \frac{\sin \omega_s' - \omega_s' \cos \omega_s'}{\sin \omega_s - \omega_s \cos \omega_s} \text{ when } \omega_s' \leqq \omega_s \tag{14}$$

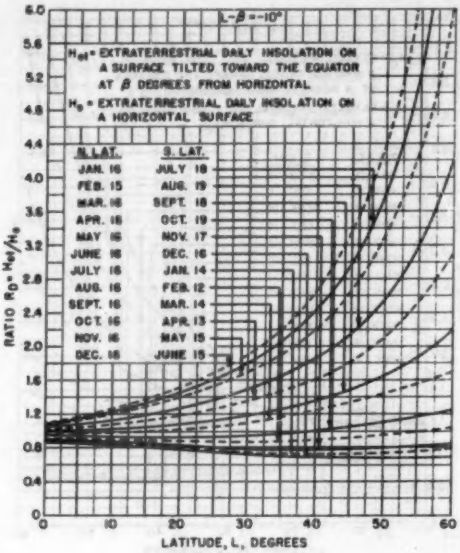


Fig. 4

Special Case 2 – Surfaces Located on the Earth and during the Time of Equinox:

In passing through the atmosphere of the earth the intensity of the direct solar radiation is reduced due to scattering and/or absorption by water vapor, clouds, dust particles, ozone, dry air molecules, etc. Consequently, on the surface of the earth the observed intensity of the direct solar radiation is only a fraction, τ, of the intensity of solar radiation incident on top of the atmosphere. Hence, by definition,

$$R_D = \frac{\displaystyle\int_{-(\omega_s \text{ or } \omega_s')}^{\omega_s \text{ or } \omega_s'} \tau\, I_{on}\left[\cos(L-\beta)\cos\delta\cos\omega + \sin(L-\beta)\sin\delta\right] d\left(\frac{24}{2\pi}\omega\right)}{\displaystyle\int_{-\omega_s}^{\omega_s} \tau\, I_{on}\left[\cos L\cos\delta\cos\omega + \sin L\sin\delta\right] d\left(\frac{24}{2\pi}\omega\right)} \quad (15)$$

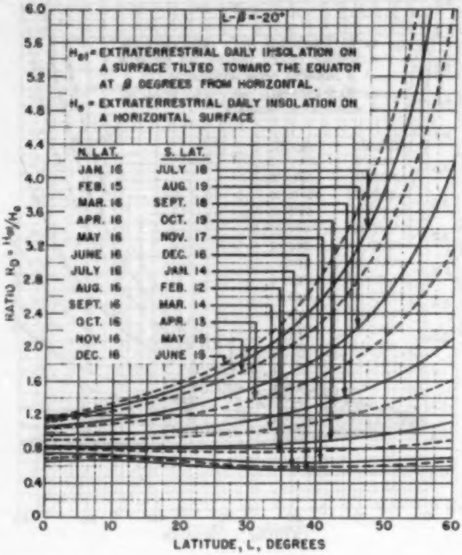


Fig. 5

where for any specific day, τ is a function of the hour angle and in general R_D can be evaluated only when this functional relationship is known. However, on equinox when $\delta = 0$ and $\omega_s = \omega_s' = \frac{\pi}{2}$.

$$R_D = \frac{\cos (L - \beta) \displaystyle\int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} \tau \cos \omega \, d\omega}{\cos L \displaystyle\int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} \tau \cos \omega \, d\omega} = \frac{\cos (L - \beta)}{\cos L} \qquad (16)$$

which is completely independent of how the values of τ may vary with the variation of the atmospheric cloudiness, water vapor, etc., during the day. It should be noticed also that the expression for R_D in Equation (16) is precisely the same expression given by Equations (13) and (14) when $\delta = 0$.

The two special cases considered above show that although Equations (13) and (14) are derived by considering the extraterrestrial solar radiation only, they do give

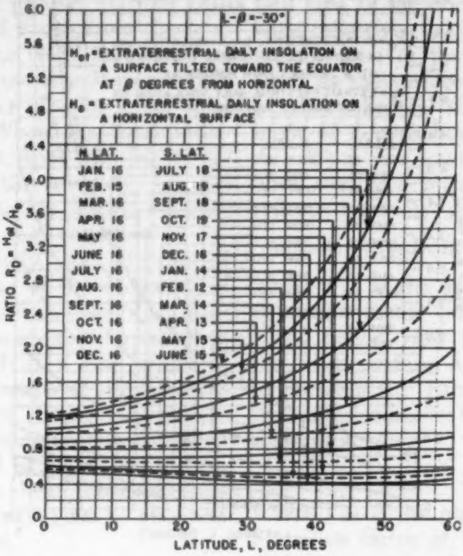


Fig. 6

the correct expressions for R_D on equinoxes for surfaces located on the earth and tilted toward the equator at any angle β. Since the exact evaluation of R_D by means of Equation (15) requires a knowledge of the atmospheric transmission coefficient, τ, which except for cloudless days is now only meagerly understood, it is recommended Equations (13) and (14) be used as a first approximation.

It should be clear that except during the time of equinoxes the application of these equations should be made on a long term statistical average basis only, since the extremely variable cloudiness would preclude the possibility of predicting the value of R_D for any single day that is not completely cloudless. It should also be expected that the use of these equations will give best results when the solar declination is not large and probably becomes less reliable for times during the solstices. Fortunately, a set of experimental data for a south facing vertical surface is available for Blue Hill, Massachusetts, and the overall error involved with the approximations here made can be determined when the theoretically computed conversion factors are compared with those derived from the experimental data.

The values of R_D are computed for each of the twelve months using the mean solar declination

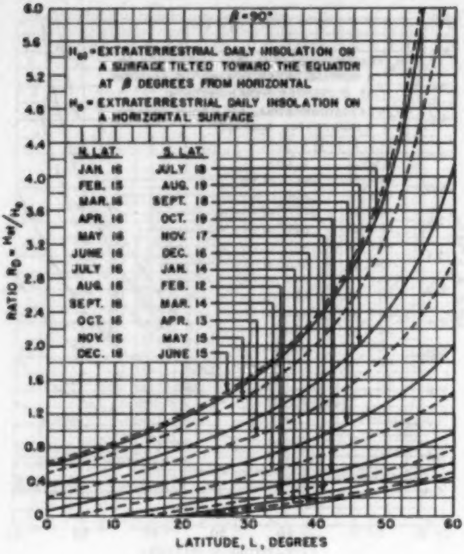


Fig. 7

and are shown in Figs. 3 to 7 for surfaces tilted at angles respectively equal to and 10, 20, and 30 deg larger than the latitude and for the special case of $\beta = 90$ deg (vertical surface).

2. Conversion factor for diffuse sky radiation – It is generally observed that diffuse radiation not only varies in intensity with the atmospheric water vapor, clouds, dusts, contents, but also in angular distribution when the atmospheric condition changes. If it is assumed as an approximation that the diffuse sky radiation is isotropic, i.e. it is uniform in all directions, then it can be shown that the ratio of the intensity of diffuse radiation incident upon a surface tilted at β deg from the horizontal surface to the intensity of diffuse radiation incident upon a horizontal surface is $\frac{1}{2}(1 + \cos\beta)$. Since this ratio is independent of the position of the sun in the sky, it also represents the ratios of the hourly and daily sums of diffuse radiation on a tilted surface to those on a horizontal surface. Thus

$$R_d = \tfrac{1}{2}(1 + \cos\beta) \tag{17}$$

3. Conversion factor for the ground reflected radiation – Since the majority of the ground surfaces normally encountered, such as grass, concrete and sand, reflect radiation more or less diffusely, only the case of perfect diffuse reflection will be considered.

Let the ground in front of a surface tilted β deg from the horizontal surface be of infinite extent and have a uniform hemispherical reflectivity (or albedo) ρ, for solar radiation. When the total radiation incident upon the horizontal surface has the intensity, I_{Th}, the reflected radiation has the intensity ρI_{Th}. Since the radiation is reflected diffusely, the problem is similar to one of the exchange of thermal radiation when the horizontal surface is emitting radiation diffusely and at a rate equal to ρI_{Th}. The intensity of the radiation incident upon the tilted surface due to ground reflection can then be computed by methods of thermal radiation heat transfer.[4] The following equation can be obtained by integration,

$$I_\rho = \frac{(1 - \cos\beta)}{2} \rho I_{Th} \qquad (18)$$

where

I_ρ = intensity of radiation reflected onto the tilted surface from the ground, Btu/hr/sq ft

Thus the conversion factor for ground reflected radiation is

$$R_\rho = \frac{(1 - \cos\beta)}{2} \rho \qquad (19)$$

When the ground is not of uniform reflectivity, the conversion factor can be determined in the following manner. It will be assumed that the ground surface can be subdivided into a finite number of portions so that a single hemispherical reflectivity can be assigned to each portion. Let the surface area of the portion of ground with reflectivity ρ_i be A_i and the surface area of the tilted surface (such as the surface of a window, a solar collector, the sensing element of a pyrheliometer, etc.) be A_s. Then the radiation incident upon the tilted surface due to the reflection of A_i is

$$q_i = A_i F_{i-s} \rho_i I_{Th} \text{ Btu/hr} \qquad (20)$$

where F_{i-s} is the geometrical configuration factor (or angle factor, shape factor, view factor, etc.) of the titled surface A_s with respect to the area A_i. However, for perfect diffuse surfaces this reciprocity relation is true,

$$A_i F_{i-s} = A_s F_{s-i} \qquad (21)$$

where F_{s-i} is the geometrical configuration factor of the surface A_i with respect to the surface A_s. Combining Equations (20) and (21)

$$q_i = A_s F_{s-i} \rho_i I_{Th} \qquad (22)$$

The average radiation intensity on the tilted surface due to the reflection of the entire ground surface is given by

$$I_\rho = (\Sigma_i q_i)/A_s = I_{Th} \Sigma_i F_{s-i} \rho_i \qquad (23)$$

Thus for the case of non-uniform reflectivity,

$$R_\rho = \Sigma_i F_{s-i} \rho_i \qquad (24)$$

Since the conversion factor is given by Equation (19) for a ground surface of uniform reflectivity, an average hemispherical reflectivity ρ for the ground of non-uniform reflectivity can be defined by equating Equations (19) and (24),

$$\rho = 2 (\Sigma_i F_{s-i} \rho_i)/(1 - \cos\beta) \qquad (25)$$

With this definition of ρ the con-

[4] Eckert, E. R. G. and Drake, R. M., Jr., "Heat and Mass Transfer," McGraw-Hill Book Company, 1959.

version factor for the ground reflected radiation for either of the above considered cases is given by Equation (19).

The values of the reflectivity (or albedo) of various types of surfaces obtained by different observers have been compiled by List[5] and part of his compilation is reproduced here in Table I.

4. Diffuse sky radiation—Since the most commonly available insolation data are the total radiation, H, and the computation of the insolation on a tilted surface by means of the method here proposed requires also the diffuse component, D, the accuracy to which H_t can be computed in many cases depends largely on the accuracy to which the diffuse radiation can be estimated. It has been shown by the authors[6] that when statistical averages are considered, relations exist between the diffuse and total radiation on a horizontal surface. These relations, reproduced here in Fig. 8, can be utilized for the estimation of the diffuse radiation when the total radiation is known.

In Fig. 8, Curve (1) gives the relation between D and H, and Curve (2) gives the relation between $\overline{D}$ and $\overline{H}$, where $\overline{D}$ and $\overline{H}$ are used to denote the monthly average values. In using these curves the different meanings of the diffuse radiation D and $\overline{D}$ should be clearly distinguished – the former denotes the average diffuse radiation received on days when the daily total radiation, H, received is a given fraction of that incident on top of the atmosphere and therefore represents the average diffuse radiation to be expected on days of a given degree of cloudiness (the degree of cloudiness is indicated by the value of $K_T = H/H_o$), whereas the latter represents the monthly average diffuse radiation and is the average diffuse radiation for all days during the month regardless of the degree of cloudiness of the individual days.

The extraterrestrial daily insolation H_o for each month needed for the determination of K_T and $\overline{K}_T$ can be computed by means of Equation (6) with the use of the mean solar declination for each month. Fig. 9 shows H_o plotted against the latitude for each of the twelve months. These values are based upon a solar constant of 2.00 ly/min or 442 Btu/hr/sq ft.

Combining Equations (2), (17) and (19),

Table I Reflectivity of the Various Types of Surfaces

Surface	Reflectivity, %
Desert	24-28
Fields, various types	3-25
Forest, green	3-10
Grass, various conditions	14-37
Ground, bare	7-20
Mold, black	8-14
Sand, dry	18
Sand, wet	9
Snow or ice	46-86

[5] List, R. J., Smithsonian Meteorological Tables, 6th revised edition, Smithsonian Institution, Washington, D. C., Table 154 & 155, pp. 442-444, 1958.

[6] Liu, B. Y. H., and Jordan, R. C., "The Interrelationship and Characteristic Distribution of Direct, Diffuse and Total Solar Radiation." Solar Energy, Vol. 4, No. 3, July 1960.

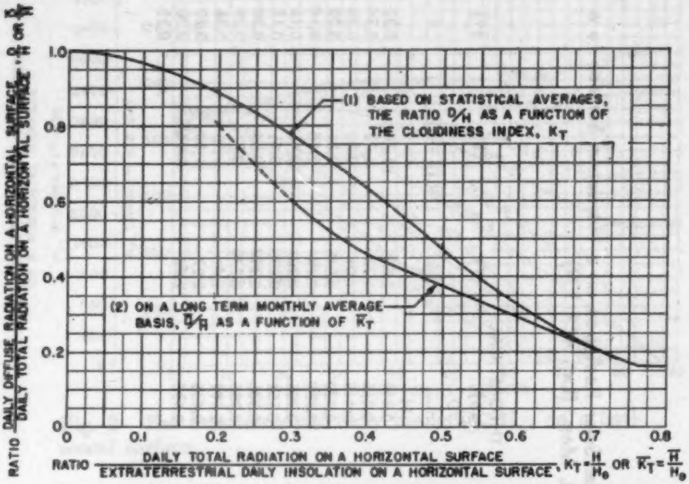


Fig. 8 The relations between the daily total and daily diffuse radiation. Curve (1) gives the ratio, D/H, of the daily diffuse to daily total radiation on a horizontal surface as a function of the cloudiness index, $K_T = H/H_o$, where D is the average daily diffuse radiation received on days when the daily total radiation, H, received is a given fraction, K_T, of the solar radiation, H_o, incident on top of the atmosphere. Curve (2) gives the ratio $\bar{D}/\bar{H}$, as a function of $\bar{K}_T = \bar{H}/\bar{H}_o$, where both $\bar{D}$ and $\bar{H}$ are respectively the long term monthly averages of the daily diffuse and daily total radiation on a horizontal surface

$$R = \left(1 - \frac{D}{H}\right) R_D + \frac{1}{2}(1 + \cos\beta)\frac{D}{H} + \frac{1}{2}(1 - \cos\beta)\rho \qquad (26)$$

COMPARISON OF RESULTS

Since 1945, the Weather Bureau of the United States has at its solar radiation station at Blue Hill, Mass., made measurements of the total radiation incident upon a south facing vertical surface and a horizontal surface. These experimental data are here used to determine the accuracy of the method here presented.

1. Conversion factor for the monthly average daily total radiation – For monthly average values, and for a vertical surface facing south, Equation (26) reduces to

Table II—Experimental and Theoretical Factors for Converting Total Solar Radiation on a Horizontal Surface to a Vertical Surface Facing South at Blue Hill, Mass. (Lat. 42°13'N)

	EXPERIMENTAL (1952-1956)			THEORETICAL					
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Month	H* langley/day	$\overline{H}_t$ langley/day	$\overline{R} = \frac{\overline{H}_t}{\overline{H}}$	H_o langley/day	$K_T = \frac{H}{H_o}$	$\frac{D}{H}$	R_D	$R\ (\rho = 0.2)$	ρ
Jan.	136	245	1.80	331	0.411	0.448	2.58	1.74	0.32
Feb.	209	289	1.38	470	0.445	0.415	1.73	1.32	0.34
Mar.	285	269	0.93	650	0.445	0.415	1.01	0.90	0.26
Apr.	368	223	0.61	836	0.440	0.420	0.515	0.61	0.20
May	462	202	0.44	962	0.481	0.389	0.281	0.47	0.14
June	534	210	0.39	1020	0.524	0.355	0.201	0.41	0.16
July	527	222	0.42	998	0.528	0.352	0.230	0.43	0.18
Aug.	431	231	0.54	888	0.485	0.386	0.397	0.54	0.20
Sept.	352	278	0.79	726	0.485	0.386	0.770	0.77	0.24
Oct.	252	311	1.23	541	0.466	0.399	1.41	1.15	0.36
Nov.	162	260	1.60	385	0.421	0.436	2.27	1.60	0.20
Dec.	125	242	1.94	296	0.422	0.435	2.88	1.94	0.20
								Avg.	0.23

* 1 langley = 3.687 Btu/sq ft

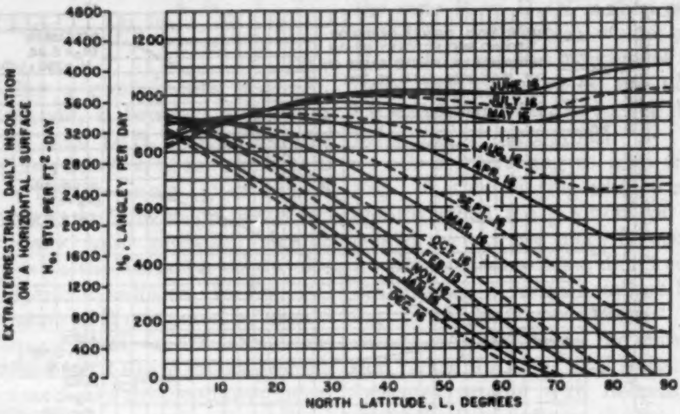


Fig. 9 The extraterrestrial daily insolation, H_o, on a horizontal surface

$$R = H_t/\bar{H} = \left(1 - \frac{\bar{D}}{\bar{H}}\right) R_D + \frac{1}{2}\frac{\bar{D}}{\bar{H}} + \frac{1}{2}\rho \qquad (27)$$

In Table II, Columns (1) to (3) show the radiation and conversion factors derived from the five-year (1952-1956) data for Blue Hill, Mass.,[7] and Columns (4) to (8) show the various quantities for the computation of the theoretical conversion factors. The theoretical conversion factors, $\bar{R}$, in Column (8) have been computed using an arbitrarily chosen reflectivity ρ of 0.2 for the ground surface in front of the vertical south facing pyrheliometer at Blue Hill. However, with the actual conversion factors known from the experimental data, Equation (27) can be used to solve for the correct "reflectivities" which will make the theoretical and experimentally determined conversion factors agree exactly. The "reflectivities" so computed are shown in Column (9).

It is doubtful that these are the actual values of the reflectivity of the ground surface at Blue Hill, since they also include experimental error and errors arising from the various approximations here made, and the authors can make no direct comparison since the actual reflectivities are unknown. However, the fact that these reflectivities are of the correct order of magnitude indicates that the method here proposed is valid. The abrupt increase of the value of ρ during January and February is expected as a result of

[7] Blue Hill solar radiation data obtained from Mr. C. V. Cunniff, U. S. Weather Bureau, Blue Hill Observatory, Milton, Mass.

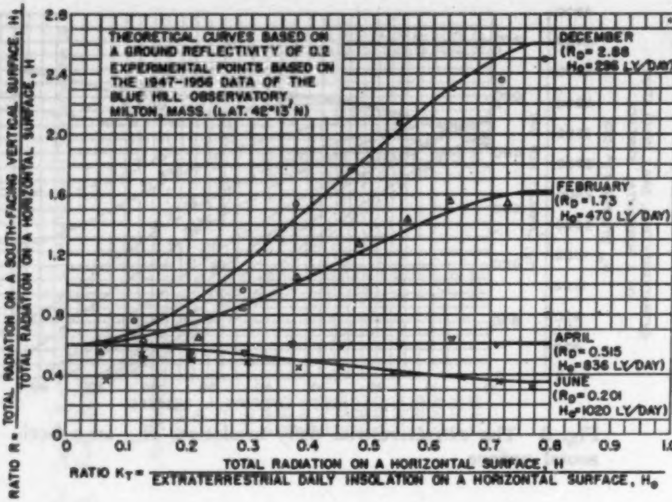


Fig. 10 The theoretical and experimental factors for converting the daily total radiation from a horizontal surface to a vertical surface facing south at Blue Hill, Mass., showing the dependence on the atmospheric cloudiness

the presence of snow cover on the ground.

The effect of ground reflectivity on the insolation received on a vertical surface facing south can be seen also from Table II. For example, for January, if after a fresh snow fall, the reflectivity of the ground is increased from 0.2 to 0.8 the conversion factor is increased from 1.74 to 2.04 – an increase of 17%. Therefore for winter collection of solar energy by means of vertical collectors, the snow cover in front of the collectors should be preserved.

On the other hand, for the month of June, when the reflectivity is increased by 0.15 (an increase easily achieved if the ground is changed from black with 10% reflectivity to grass of 25% reflectivity), the radiation incident upon the south facing vertical surface is increased approximately 20%. If other things remain equal, this should be avoided for summer air conditioning.

2. Variation of the conversion factor for daily total radiation during the same month with the degree of atmospheric cloudiness – For days of a given degree of atmospheric cloudiness, and for a vertical surface facing south, Equation (26) becomes,

$$R = (1 - \frac{D}{H}) R_D + \frac{1}{2}\frac{D}{H} + \frac{1}{2}\rho \quad (28)$$

For a given month when R_D is constant, R becomes a function of the ratio D/H. Since the ratio D/H is a function of the cloudiness index K_T, as Curve (1) of Fig. 8 indicates, the conversion factor for the daily total radiation should vary with the variation of the atmospheric cloudiness. In Fig. 10, the values of R computed by means of Equation (28) for the latitude of Blue Hill (42° 13′) are shown plotted as a function of the cloudiness index K_T for the four months: December, February, April and June. The experimental points shown are derived from the ten year (1947-1956) data for Blue Hill.

The experimental conversion factors for each month have been obtained by first classifying the days into a number of groups such that the values of H for days belonging to each group fall within a limited interval of values (the sizes of the intervals are indicated by the differences of the values of H of the neighboring points) and the ratio $R = H_t/H$ is then computed for each group where both H_t and H are the average of the daily total radiation on the tilted and horizontal surfaces for the days belonging to each group.

It is interesting to note from Fig. 10 that during the winter months when the positions of the sun in the sky are more favorable for the incidence of the direct solar radiation on a vertical surface facing south and consequently the values of R_D are large, the conversion factor for daily total radiation increases rapidly with decreasing atmospheric cloudiness due to the decrease of the ratio D/H. On the contrary, for the month of June, when R_D is small, the value of R decreases slightly when K_T increases. This does not mean, however, that during the month of June the radiation incident upon a south facing vertical surface at Blue Hill becomes less as the atmosphere becomes clearer. Because the radiation incident upon the vertical surface is the product RH and is not determined by the value of R alone.

DISCUSSION

E. C. MILES, Pittsburgh, Pa.: How are the data obtained from these computations compared with the data from different latitudes and hours of the day as given in the GUIDE?

AUTHOR LIU: The values given in the GUIDE are for clear days only and are tabulated for different solar altitude angles and hours of day. This paper is concerned with the daily radiation for all days; therefore, no direct comparison with the values given in the GUIDE can be made.

1763

No. 1763

Calculated Temperature Rise in Round Ducts

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Residential summer cooling applications frequently require installation of ductwork in non-air conditioned spaces such as basements, crawl spaces and attics. The conditioned air within the ducts is subject to heat gain from the non-conditioned spaces, since the ambient temperature may vary from 75 F in basements to 140 F or more in attic spaces.

An experimental investigation of heat gain to air flowing in ducts requires that the ducts be enclosed in a space throughout which a uniform air temperature may be maintained. Considerable expense would be involved to provide such a space in a laboratory, plus the controls necessary to maintain the proper conditions within the space. As an alternate means of studying the problem, the ductwork in the basement and attic of Warm Air Heating Research Residence No. 2[1] was provided with sufficient instrumentation to measure the average temperature rise of the conditioned air.

It should be noted that the instrumentation was not considered suitable for determination of heat gain by the differences in enthalpies at two stations in a duct. This would require an evaluation of temperature and velocity profiles to determine total enthalpies at each station. The Residence installation did not provide a means of controlling the ambient air temperature of the basement and attic. However, it was possible to measure the average temperature rise in the ducts after the summer air conditioner had been in operation for several hours and steady-state conditions prevailed within the basement or attic.

Several years ago, in an investigation[2] of temperature drop in

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J. Richard Wright is an Assistant Professor of Mechanical Engineering at the Tennessee Polytechnic Institute. Edward J. Brown is a Research Assistant Professor of Mechanical Engineering at the University of Illinois. This paper was prepared for presentation at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961.

ducts conveying heated air, it was shown that measurements of heat transfer coefficients (made in the Mechanical Engineering Laboratory of the University of Illinois) were in agreement with the values for the same coefficients published in the literature. Therefore, it was possible to calculate accurately the temperature drop in ducts used in forced warm air heating systems. It was assumed that similar calculations of temperature rise in ducts conveying cooled air would be valid. The measured average temperature rise data from the Residence, for a necessarily limited number of ambient temperatures, compared favorably with calculated values of temperature rise for the same ambient temperatures.

The purpose of this paper is to present an analysis of the effects of duct air temperature and velocity and ambient air temperature on the temperature rise of air flowing in round ducts used for summer air conditioning. The analysis is based on the following assumptions: (1) The heat transfer rate is constant; does not vary with time, (2) The mean radiant temperature is equal to the ambient air temperature and (3) Condensation does not occur on the exterior surface of uninsulated ducts or on the exterior vapor barrier surface of insulated ducts.

Uninsulated Ducts—Heat transfer, from the surroundings to the conditioned air flowing in a duct, depends upon the heat transfer coefficients of free convection and radiation at the outside surface of the duct and the coefficient of heat transfer of forced convection at the inside surface of the duct. The resistance to heat flow of the duct wall has been neglected because of the small thickness of the wall and the relatively large thermal conductivity of the sheet metal used in the manufacture of ducts. The heat transfer at the inside and outside surfaces must be equal when steady-state heat transfer exists. The heat balance can be expressed by the following equation:

$$q = h_i A (t_w - t_M) = (h_r + h_c) A (t_a - t_w) \quad (1)$$

where q = heat transfer, Btu/hr

h_i = inside surface coefficient of forced convection heat transfer, Btu/hr/sq ft/F

t_w = duct surface temperature, F

t_M = mean temperature of conditioned air, F

h_c = outside surface coefficient of natural convection heat transfer, Btu/hr/sq ft/F

h_r = outside surface coefficient of radiant heat transfer, Btu/hr/sq ft/F

A = area of duct surface, assumed equal for inside and outside surface because of small wall thickness, sq ft

t_a = ambient temperature, F

The temperature change of the air flowing in the duct is not a linear function of duct length, but for short sections of duct the change may be considered linear. It was assumed that the temperature change would be approximately linear in a 10-ft length; therefore, the area referred to is the wall area of 10 linear ft of duct.

The solution of Equation (1) requires an evaluation of the heat transfer coefficients for assumed ambient and duct air temperatures. The inside surface coefficient of

heat transfer by forced convection was evaluated by the equation for forced convection in turbulent flow in horizontal pipes found in most heat transfer texts:[3]

$$h_i = 0.023 \frac{k}{D} \left[\frac{V D \rho}{\mu}\right]^{0.8} \left[\frac{c \mu}{k}\right]^{0.4} \quad (2)$$

where

k = thermal conductivity of the fluid (air), Btu/hr/sq ft/F

D = diameter of duct, ft

V = average velocity of fluid (air), ft/sec

ρ = fluid density, lb/cu ft, standard conditions

μ = dynamic viscosity of fluid, lb/sec/ft

c = specific heat of fluid, assumed constant, 0.24 Btu/lb/F

$\frac{c\mu}{k}$ = Prandtl number

$\frac{VD\rho}{\mu}$ = Reynold's number

The properties of air were evaluated at the mean duct air temperature. The values of thermal conductivity were obtained from Gas Tables[4] by Keenan and Kaye. Over the range of temperatures from 32 to 122 F, the Prandtl number varied from 0.712 to 0.701 and an average value of 0.706 was used throughout the calculations. The inside surface coefficients of heat transfer were evaluated for duct air velocities from 400 to 1200 fpm in increments of 200 fpm, and mean duct air temperatures of 50 to 70 F in increments of 5 F for 4, 5, 6, 7 and 8-in. diam ducts.

The outside surface coefficient of heat transfer due to natural convection was determined by the equation for natural convection found in most heat transfer texts:[3]

$$h_c = 0.53 \frac{k}{D} \left[\frac{D^3 \beta \rho^2 g \, (t_A - t_W)}{\mu^2}\right]^{0.25} \left[\frac{c \mu}{k}\right]^{0.25} \quad (3)$$

where β = temperature function, reciprocal of absolute temperature, 1/(deg Rankine)

g = gravitational acceleration, 32.16 ft/sec²

For the determination of the coefficient of natural convection, the film temperature (the temperature of the thin stagnant layer of air adjacent to the surface) was assumed equal to the average of the ambient air and duct surface temperatures; the properties of the air film, thermal conductivity, density, dynamic viscosity and the temperature function, were evaluated at the film temperature. The coefficients were determined for ambient temperatures of 75, 100, 125 and 150 F and duct surface temperatures of 55 to 100 F in 5 F increments.

The outside surface coefficient of radiant heat transfer was evaluated by the Stefan-Boltzmann[3] equation divided by the difference in ambient and duct surface temperatures, assuming a configuration factor of unity for a small duct located in a relatively large enclosure:

$$h_r = \frac{\sigma \epsilon \, (T^4_A - T^4_W)}{(T_A - T_W)} \quad (4)$$

where

σ = Stefan-Boltzmann constant, 0.173×10^{-8} Btu/hr/sq ft/F^4

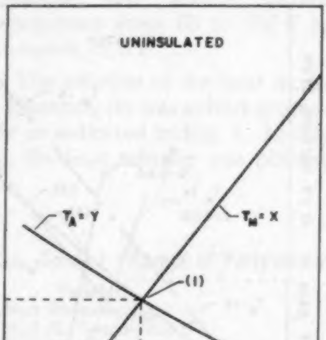
ε = surface emissivity of duct

T_A, T_W = ambient temperature and duct surface temperature, deg R

Equations (1) through (4) apply to any thin wall duct. For the purpose of this analysis, only gray oxidized zinc surfaces with a surface emissivity $\varepsilon = 0.25$ were considered. The ducts installed in the Residence were also of galvanized iron. Radiation coefficients were determined for the same range of ambient and duct surface temperatures as the natural convection coefficients.

After the coefficients of heat transfer were determined, the graphical method illustrated in Fig. 1 was used to solve Equation (1) for the heat transfer rate and duct surface temperature. In Fig. 1 each side of Equation (1) is plotted as a function of the duct surface temperature, t_W. The left-hand side of Equation (1) represents the heat transfer through the inside surface. It is represented by a line designated $t_M = x$. For a constant duct air temperature, the heat transfer increases almost linearly with duct surface temperature and is zero for $t_M = t_W$. The heat transfer to the outside surface of the duct at a constant ambient temperature, $t_A = y$, is shown. As the duct surface temperature, t_W, increases, the temperature difference decreases; therefore, the heat transfer would decrease to zero at $t_A = t_W$.

Fig. 1 Graphical solution of equation for heat transfer to air flowing in uninsulated ducts



For a specified condition of duct air velocity, duct air temperature and ambient temperature, the intersection (1) of the two curves provides a solution of Equation (1). The duct surface temperature and the heat transfer rate are obtained directly from the coordinates.

The temperature rise in a 10-ft length of duct was determined from the equation:

$$\Delta t = \frac{q}{60\, V\, A\, \rho\, c} \tag{5}$$

where

Δt = temperature rise of air, F/10 lineal ft

60 = factor to convert velocity from ft/min to ft/hr

A = cross-sectional area of duct, sq ft

The duct surface temperature determines whether condensation will occur. Condensation of vapor will occur if the duct surface tem-

perature is below the dew point of the ambient air. The dew point is a function of the ambient air dry-bulb temperature and relative humidity.

The mean duct air temperature, t_M, represents the temperature at the mid-length of a 10-ft section of duct. The entering air temperature is obtained by subtracting one-half Δ^t from the mean temperature:

$$t_i = t_M - \frac{\Delta_t}{2} \qquad (6)$$

where t_i = duct air temperature at the entrance of the 10-ft section. The temperature rise was plotted as a function of entering temperature for the various duct air velocities and ambient temperatures. These graphs are discussed in a later section.

Insulated Ducts – The analysis for insulated ducts is similar to the analysis for uninsulated ducts except that the resistance to heat flow imposed by the insulation must be considered. The heat transfer through the inside surface of the duct due to forced convection is the same as for uninsulated ducts. The heat transfer to the outer surface by natural convection and radiation is the same as for uninsulated ducts, except that the outside surface temperature, t_S, is substituted for the duct surface temperature, t_W, and the outside surface area, A_S, is substituted for the duct surface area, A.

The heat transfer through the insulation is given by the conduction equation:

$$q = C\,A_M\,(t_s - t_w) \qquad (7)$$

where

C = thermal conductance of the insulation, Btu/hr/sq ft/F

A_M = log mean area of insulation, $\dfrac{A_s - A_w}{\ln \dfrac{A_S}{A_w}}$, sq ft.

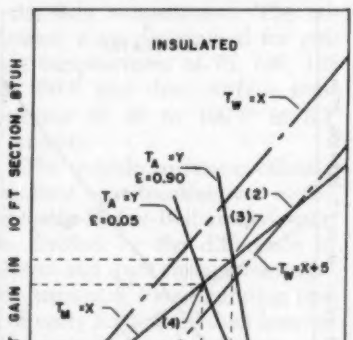
Setting the heat transfer through each resistance equal:

$$q = h_i\,A_w\,(t_W - t_M) = C\,A_M\,(t_s - t_W) = (h_c + h_r)\,A_s\,(t_A - t_s) \qquad (8)$$

The inside surface coefficient of heat transfer by forced convection and the heat transfer rate were the same as evaluated for uninsulated ducts.

The coefficient of natural convection was calculated by a procedure identical with that for the

Fig. 2 Graphical solution of equation for heat transfer to air flowing in insulated ducts



coefficient for uninsulated ducts, except that the diameter used in Equation (3) was the outer diameter of the insulation. The surface temperature was t_S instead of t_W.

The radiation coefficient was also calculated similarly to the coefficient for uninsulated ducts. Calculations were made for two types of vapor barriers, aluminum foil and duplex paper. An emissivity of 0.05 was assumed for aluminum foil and an emissivity of 0.90 was assumed for duplex paper, which consisted of two layers of Kraft paper bonded together with a thin layer of asphalt. The conductivity of the insulation was assumed to be 0.27 Btu/hr/sq ft/F/in. of thickness, which gave conductance of 0.27 Btu/hr/sq ft/F for 1-in. thick insulation and 0.135 Btu/hr/sq ft/F for 2-in. thick insulation. Calculations of the heat transfer rate through the insulation were made for both thicknesses of insulation, for duct surface temperatures from 55 to 75 F in 5 F increments and outside surface temperatures from 55 to 150 F in increments of 5 F.

The solution of the heat transfer Equation (8) was solved graphically as indicated in Fig. 2. In this case, the heat transfer was plotted as a function of the outside surface temperature, t_S.

The equation of heat transfer through the insulation yields lines of constant duct wall temperature represented by the lines $t_W = x$ and $t_W = x + 5$. The heat transfer, q, is zero when $t_W = t_S$ and increases linearly as t_S increases, for a constant duct surface temperature. For a constant outside surface temperature the heat transfer rate decreases as the duct surface temperature increases, therefore, the line for $t_W = x + 5$ is below the line for $t_W = x$.

The heat transfer to the outer insulation surface is represented by the two lines designated $t_A = y$. The upper line is for the aluminum foil vapor barrier (emissivity = 0.05) and the lower line is for the duplex paper vapor barrier (emissivity = 0.90). The effectiveness of the lower emissivity vapor barrier in reducing the heat transfer rate and temperature rise is apparent.

The heat transfer through the inside film is dependent upon the mean duct air temperature and the duct surface temperature. If $t_M = t_W = x$, for example, the heat transfer rate is zero as represented by intersection (1) on Fig. 2. If $t_W =$

Table I Range of Variables in Analysis of Heat Gain to Ducts

Variable	Range
Duct Diameter, in.	4, 5, 6, 7, 8
Duct Air Temperature, F	50, 55, 60, 65, 70
Duct Air Velocity, fpm	400, 600, 800, 1000, 1200
Ambient Temperature, F	75, 100, 125, 150
Insulation Thickness, in.	0, 1, 2
Vapor Barrier Emissivity	0.25 for uninsulated 0.05 and 0.90 for insulated

Table II Temperature Rise in 10-ft Sections of Insulated Ducts, F/ft
75 F Ambient Air

Entering Air Temperature, F	4-in. Diam.				5-in. Diam.				6-in. Diam.				7-in. Diam.				8-in. Diam.			
Vapor Barrier Surface Emissivity	0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90	
Insulation Thickness, in.	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2
50 F	.07	.05	.09	.06	.06	.04	.07	.04	.05	.03	.05	.03	.04	.03	.05	.03	.03	.02	.04	.02
55 F	.06	.04	.07	.05	.04	.03	.05	.04	.04	.02	.04	.03	.03	.02	.04	.02	.03	.02	.03	.02
60 F	.05	.03	.05	.03	.03	.02	.04	.03	.03	.02	.03	.02	.02	.02	.03	.02	.02	.01	.02	.01
65 F	.03	.02	.04	.02	.02	.02	.03	.02	.02	.01	.02	.01	.02	.01	.02	.01	.01	.01	.02	.01
70 F	.02	.01	.02	.01	.01	.01	.02	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01	.01

Table III Temperature Rise in 10-ft Sections of Insulated Ducts, F/ft
125 F Ambient Air

Entering Air Temperature, F	4-in. Diam.				5-in. Diam.				6-in. Diam.				7-in. Diam.				8-in. Diam.			
Emissivity	0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90		0.05		0.90	
Insulation Thickness, in.	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2
50 F	.23	.16	.26	.17	.18	.12	.21	.13	.14	.10	.17	.10	.12	.08	.14	.09	.10	.07	.12	.07
55 F	.22	.15	.25	.16	.17	.11	.19	.12	.13	.09	.16	.10	.11	.07	.13	.08	.10	.06	.11	.07
60 F	.21	.14	.24	.15	.16	.11	.18	.12	.13	.08	.15	.09	.11	.07	.12	.08	.09	.06	.11	.07
65 F	.19	.13	.22	.14	.15	.10	.17	.11	.12	.08	.14	.09	.10	.06	.12	.07	.08	.05	.10	.06
70 F	.18	.12	.21	.13	.13	.09	.16	.10	.11	.07	.13	.08	.09	.06	.11	.07	.08	.05	.09	.06

$x + 5$ and $t_M = x$, the temperature potential is 5 F and the heat transfer may be calculated for this condition; the value is represented by intersection (2) located on the $t_W = x + 5$ line. The intersection of the line of constant mean duct air temperature, t_M, with the line of constant ambient temperature, t_A, for either surface emissivity, intersection (3) or (4), provides a solution of Equation (8). The heat transfer rate and the outer surface temperature are obtained directly from the coordinates. The temperature rise and the entering temperature were determined as for uninsulated ducts, and the temperature rise was plotted as a function of entering air temperature for the various combinations.

RESULTS

Curves of temperature rise as related to entering temperature were plotted for all combinations of duct diameter, insulation thickness, duct air velocity, vapor barrier emissivity and ambient temperatures. For convenient reference the ranges of these items are listed in Table I.

Temperature Rise Curves — These, as related to entering temperature for 800 fpm velocity and 75 and 125 F ambient temperatures, are presented in Fig. 3 and Tables II and III for all duct diameters. Fig. 3 is for uninsulated ducts at 75 and 125 F ambient temperature. Table II is for 75 F ambient air and ducts with 1 and 2-in. insulation with vapor barrier surface emissivities of 0.05 and 0.90. Table III is similar to Table II but is for 125 F ambient air. The temperature rise data obtained from the figure and tables show temperature rise per lineal ft in a 10-ft section. If the temperature rise in a duct longer than 10-ft is to be determined the duct must be divided into 10-ft segments. The following example, illustrated in Fig. 3, shows the procedure.

Example 1. A 4-in. diam uninsulated duct is located in a basement where the ambient temperature is 75 F. The duct is 25 ft long. The air temperature at the entrance is 55 F and the air velocity is 800 fpm. Determine the overall air temperature rise and the exit temperature.

Solution: With the entering air temperature of 55 F, Fig. 3 indicates a temperature rise of 0.19 F/ft in the first 10-ft section. Therefore, the air temperature leaving the first 10-ft section is $55.0 + 10\ (0.19) = 56.9$ F. This is also the entering air temperature for the second 10-ft section. From Fig. 3 the temperature rise in the second 10-ft section is 0.17 F/ft which results in a leaving temperature of $56.9 + 10\ (0.17) = 58.6$ F. The entering air temperature for the last five ft of duct is also 58.6 F and the temperature rise in this section is 0.15 F/ft. The final temperature is $58.6\text{ F} + 5\ (0.15) = 59.4$ F. The overall air temperature rise is $59.4 - 55.0 = 4.4$ F.

Tables II and III permit the determination of temperature rise information for air flowing in insulated ducts with two vapor barrier surface emissivities and exposed to two ambient air temperatures. The temperature rise for unlisted entering air temperatures between 55 and 70 F may be determined by interpolation.

Velocity Correction Factors—These, based on a velocity of 800 fpm, are presented in Table IV. Information is listed for both insulated and uninsulated ducts. The factors were

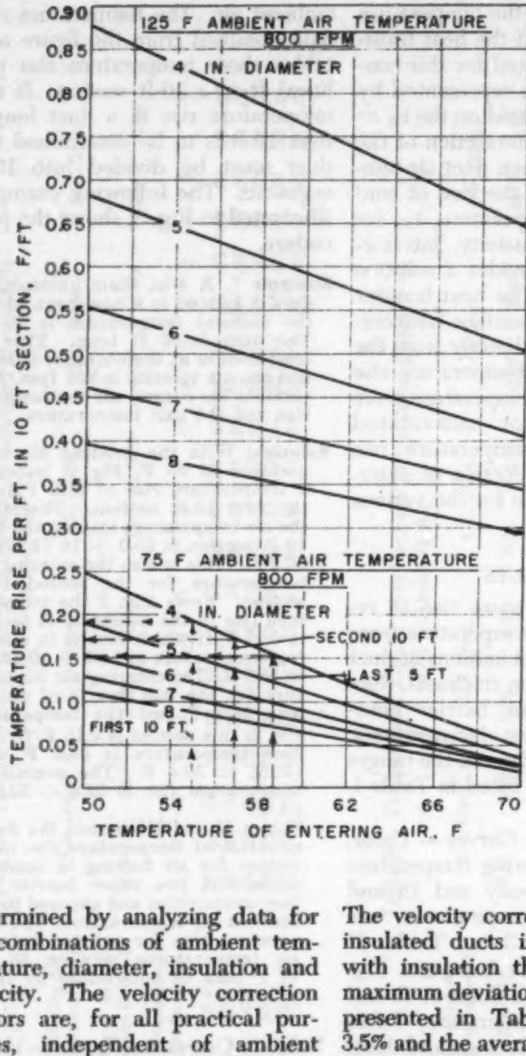


Fig. 3 Temperature rise in uninsulated ducts

determined by analyzing data for all combinations of ambient temperature, diameter, insulation and velocity. The velocity correction factors are, for all practical purposes, independent of ambient temperature and duct diameter. The maximum deviation from the values listed in Table IV was 3% and the average deviation was 2%.

The velocity correction factors for insulated ducts increased slightly with insulation thickness, but the maximum deviation from the values presented in Table IV was only 3.5% and the average deviation was 2%. The velocity correction factors permit utilization of the temperature rise data presented in Fig. 3 and Tables II and III for any

Table IV Velocity Correction Factors

Velocity, fpm	400	500	600	700	800	900	1000	1100	1200
Uninsulated Duct	1.60	1.40	1.23	1.10	1.00	0.92	0.85	0.79	0.74
Insulated Duct	1.89	1.57	1.31	1.13	1.00	0.90	0.81	0.74	0.68

velocity between 400 and 1200 fpm. To illustrate, assume that in Example 1 the velocity is 500 fpm instead of 800 fpm. The velocity correction factor for 500 fpm in uninsulated ducts is 1.40. Therefore, the temperature rise in the first 10-ft segment is 1.40 times 0.19 = 0.27 F/ft. The temperature entering the second 10-ft segment would be 55.0 + 10 (0.27) = 57.7 F.

Ambient Temperature Correction Factors—Ambient temperature correction factors are presented in Table V. The ambient temperature correction factors are based on 125 and 75 F ambient temperatures. These ambient temperatures correspond to the temperature rise data in Fig. 3 and Tables II and III. The ambient temperature correction factors are independent of velocity and diameter. They are dependent upon entering temperature and the amount of duct insulation. Presentation of the data as a function of ambient temperature would require a separate curve for each entering air temperature. However, it was found that the ratio

$$\left[\frac{\text{ambient temperature} - \text{entering temperature}}{\text{base temperature} - \text{entering temperature}}\right]$$

provided an index of the ambient temperature correction factors, and left the amount of insulation as the only remaining variable influencing the factors. Therefore, the ambient temperature correction factors are presented as a function of the ratio given above with separate values listed for uninsulated and insulated ducts.

The ambient temperature correction factors permit utilization of the temperature rise data presented in Fig. 3, and Tables II and III for any ambient temperature between 75 and 150 F. If the ambient tem-

Table V Ambient Temperature Correction Factors

100 to 150 F Ambient Range									
$\frac{T_a - T_1}{125 - T_1}$	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4
Uninsulated Duct	0.56	0.66	0.78	0.89	1.00	1.12	1.23	1.35	1.46
Insulated Duct	0.59	0.69	0.79	0.90	1.00	1.10	1.20	1.31	1.41

75 to 100 F Ambient Range									
$\frac{T_a - T_1}{75 - T_1}$	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5
Uninsulated Duct	1.63	2.25	2.90	3.50	4.15	4.80	5.45	6.10	6.80
Insulated Duct	1.50	2.00	2.50	3.05	3.55	4.05	4.60	5.10	5.60

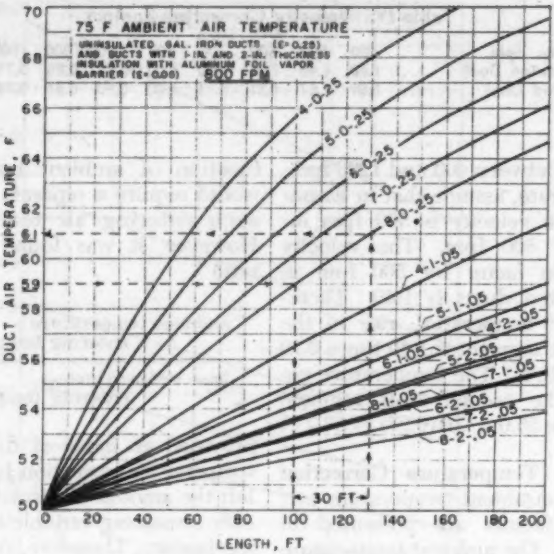


Fig. 4 Temperature-length curves for uninsulated ducts and insulated ducts with aluminum foil vapor barriers at 75 F ambient air temperature

perature is between 75 and 100 F the correction factors are based on the 75 F temperature rise data. If the ambient temperature is between 100 and 150 F the correction factors are based on the 125 F data. To illustrate, assume that in Example 1 the ambient temperature is 85 F instead of 75 F. The correction is based on the 75 F data. The ratio $\frac{t_A - t_1}{75 - t_1}$ is equal to $\frac{85 - 55}{75 - 55}$ $= 1.5$. The ambient temperature correction factor from Table V is 1.63. Therefore, the temperature rise in the first 10-ft segment of duct is 1.63 times 0.19 = 0.31 F/ft.

Utilization of Curves and Tables – The data presented in Fig. 3 and Tables II, III, IV and V are sufficient for the determination of the temperature rise occurring in most applications. The solution of a typical problem is presented below to illustrate the combined use of velocity and ambient temperature correction factors.

Example 2. A 6-in. diam duct insulated with 1-in. glass fiber insulation and covered with a paper vapor barrier (emissivity = 0.90) is located in an attic space where the ambient temperature is 140 F. The duct is 18 ft long. The air temperature at the entrance to the duct is 55 F and the air velocity is 1000 fpm.

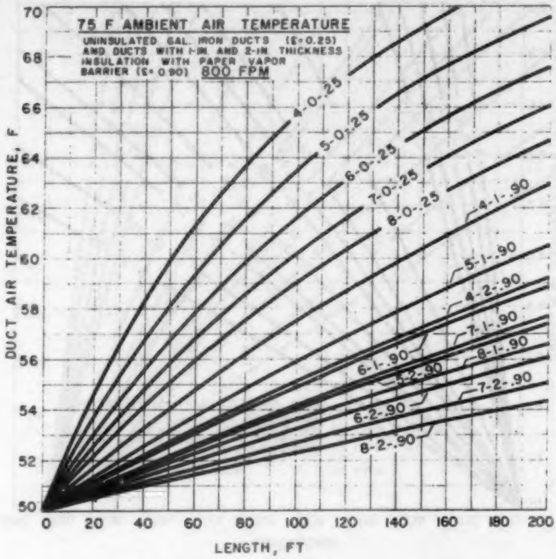


Fig. 5 Temperature-length curves for uninsulated ducts and insulated ducts with paper vapor barriers at 75 F ambient temperature

Determine the total temperature rise and the exit temperature.

Solution. **The velocity correction factor from Table IV is 0.81. From Table V the ambient temperature correction factor for a ratio $\frac{t_a - t_1}{125 - t_1}$ $= \frac{135 - 55}{125 - 55} = 1.14$ by interpolation is 1.14. From Table III for 800 fpm velocity and 125 F ambient temperature, the temperature rise in the first 10-ft segment is 0.16 F/ft. Applying the correction factors and multiplying by 10 ft, the temperature rise in the first 10 ft is (0.81) (1.14) (10) (0.16) = 1.47 or 1.5 F. The temperature entering the 8-ft segment is 56.5 F. The velocity correction factor is the same, 0.81. The new ratio must be calculated to obtain the ambient temperature correction factor:**

$$\frac{135 - 56.5}{125 - 56.5} = 1.15$$

and the ambient temperature correction obtained from Table V is by interpolation, 1.15. For 800 fpm velocity and a 125 F ambient temperature, the rise in the last segment is 0.15 F/ft. Applying the correction factors and multiplying by 8 ft, the temperature rise in the last 8 ft of the duct is (0.81) (1.15) (8) (0.15) = 1.1 F. The exit temperature is 56.5 + 1.1 = 57.6 F and the total temperature rise is 2.6 F.

Temperature-Length Curves – Determination of the temperature rise in ducts is more convenient when the data are presented in the form of temperature-length curves as

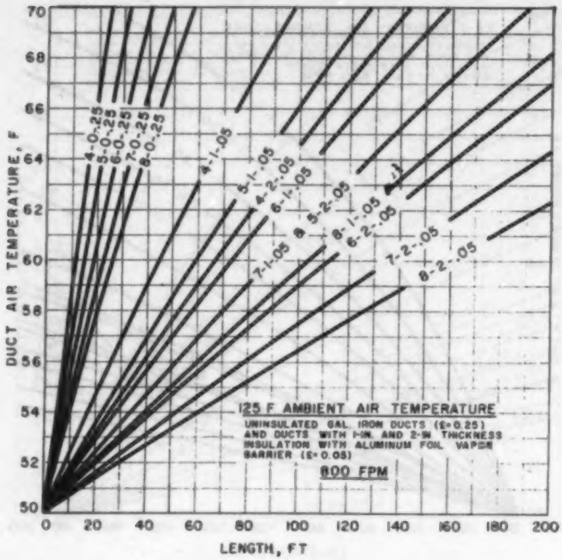


Fig. 6 Temperature-length curves for uninsulated ducts and insulated ducts covered with aluminum foil vapor barrier at 125 F ambient air temperature

shown in Figs. 4 through 7. These curves were developed from the temperature rise data presented in Fig. 3 and Tables II and III and are valid for 800 fpm velocity and either 75 or 125 F ambient temperature. Figs. 4 and 5 are for 75 F ambient temperature and Figs. 6 and 7 are for 125 F ambient temperature. Figs. 4 and 6 are for uninsulated ducts and insulated ducts with aluminum foil vapor barrier ($\varepsilon = 0.05$). Figs. 5 and 7 repeat the data for uninsulated ducts and contain data for insulated ducts with paper vapor barrier ($\varepsilon = 0.90$). The duct diameter, insulation thickness and exterior surface emissivity is indicated for each curve. For example, 8-1-.05 indicates an 8-in. diam duct with 1-in. insulation and an aluminum foil vapor barrier (emissivity = 0.05). The emissivity of a paper vapor barrier is 0.90 and the surface emissivity of a galvanized iron duct is 0.25. The following example, illustrated in Fig. 4, shows how the temperature-length curves may be utilized.

Example 3. Assume that 59 F air enters an 8-in. diam galvanized iron duct which is uninsulated and 30 ft in length. The ambient air temperature is 75 F.

The arrows on Fig. 4 indicate the

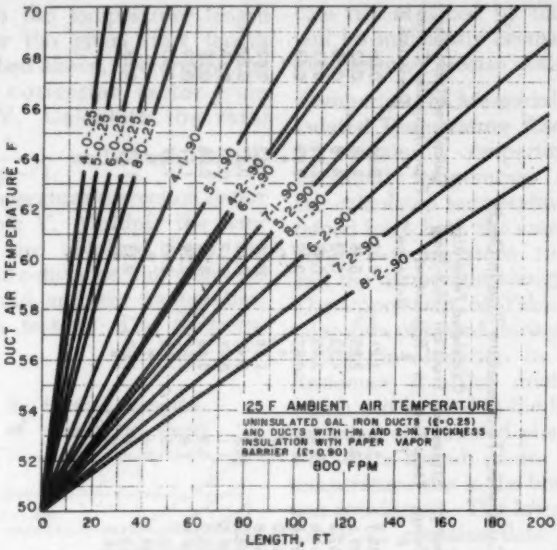


Fig. 7 Temperature-length curves for uninsulated and insulated ducts covered with paper vapor barriers at 125 F ambient air temperature

use of the curves to determine the final temperature of air leaving the duct. Enter the chart on the ordinate at 59 F (entering temperature). Proceed horizontally to the duct specifications curve (8-0-.25 curve for this example). From the intersection with the curve proceed downward to the length scale on the abscissa (100 ft in this case). Add the length of the duct to the intersected length (100 + 30 = 130 ft). Proceed upward on the 130 ft length line and again intersect the 8-0-.25 curve. From this intersection proceed to the left to the ordinate and read the leaving temperature (61 F in this case).

Ambient temperature and velocity correction factors are not strictly applicable to temperature-length curves, but if the ducts are less than 50 ft long the error introduced will be small; less than 5% for insulated ducts and less than 20% for uninsulated ducts. Larger errors occur for smaller ducts and lower velocities. Specific examples of the variation between the temperature rise data obtained by temperature-length curves for other than 800 fpm velocity and 75 to 125 F ambient temperatures are included in the following section.

Adjustments to other than the base ambient temperatures and velocity are made in the following manner: Obtain the temperature

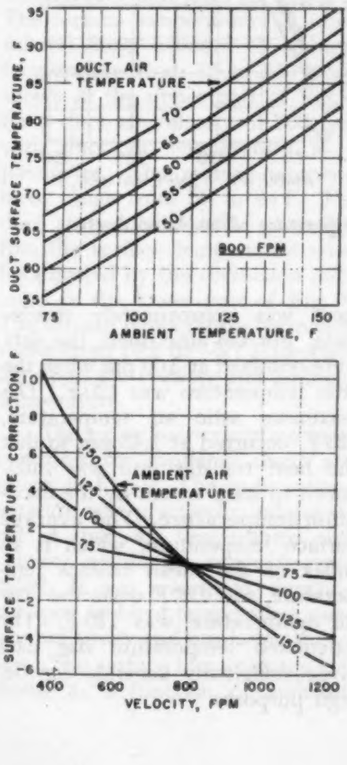
Table VI Comparison of Measured and Calculated Temperature Rise in Ducts

Location	Ambient Air Temp, F	Duct Diam, in.	Surface Emissivity	Length, ft	Vel., fpm	Entering Air Temp, F	Measured Temp Rise, F	Calculated Temp Rise From Ent Temp-Temp Rise Curves F	Calculated Temp Rise From Ent Temp-Temp Rise Curves Deviation, F	Calculated Temp Rise From Temp-Dist Curves F	Calculated Temp Rise From Temp-Dist Curves Deviation, F
		4	0.25	16	1060	58.9	2.8	2.81	0.01	2.84	0.04
		4	0.25	16	1040	59.6	3.5	2.85	0.65	2.88	0.62
		4	0.25	16	410	62.3	3.6	4.33	0.73	4.18	0.58
Basement	78.5	4	0.25	11	620	61.6	2.0	2.45	0.45	2.51	0.51
		4	0.25	4	720	59.6	0.4	0.82	0.42	0.83	0.43
		4	0.25	11	340	62.0	4.3	*	*	*	*
		4	0.25	7	870	56.8	1.6	1.54	0.06	1.62	0.02
		4	0.25	3	1100	58.5	0.3	0.47	0.17	0.57	0.27
		7	0.05	10	546	58.5	1.1	0.93	0.17	0.93	0.17
		7	0.90	8	669	58.7	0.9	0.68	0.22	0.68	0.22
		7	0.05	15	501	58.5	1.8	1.51	0.29	1.51	0.29
Attic	120.0	7	0.90	17	484	58.7	2.5	1.85	0.65	1.97	0.53
		6	0.90	4½	257	59.2	0.4	*	*	*	*
		6	0.05	11	640	58.2	1.4	1.07	0.33	1.12	0.28
		5	0.90	13	278	58.5	5.5	*	*	*	*

* The duct air velocity was below the range covered in the calculations.

rise from the temperature-length curve for the given duct length as described above. Determine the velocity correction factor from Table IV. Calculate the ratio $\frac{t_A - t_1}{\text{base temp} - t_1}$ and obtain the ambient temperature correction factor from Table V. Multiply the temperature rise obtained from the temperature-distance curve by the velocity and ambient temperature correction factors. The temperature rise obtained by this method will be sufficiently accurate for all but the most precise designs.

Fig. 8 Surface temperature of uninsulated ducts



Comparison of Measured and Calculated Temperature Rise – Table VI contains a comparison of the measured temperature rise with the calculated temperature rise obtained from both the entering temperature-temperature rise curves and the temperature-length curves. The upper part of Table VI contains data obtained during the 1954 cooling investigation in Research Residence 2. The duct system, which was located in the basement, consisted of extended plenums and 4-in. diam branch ducts. Only the temperature rise in the branch duct was considered. The lower part of Table VI contains data obtained during the 1957 cooling investigation in Research Residence 2.

The duct system consisted of a central plenum and individual branch ducts to each ceiling diffuser. The calculated data were corrected to the attic ambient temperature and the velocity in each duct. The maximum deviation between the measured data and the calculated data was 0.73 F. The average deviation was 0.33 F and the deviation was almost the same, regardless of which method was used in the calculations.

The maximum deviation between the temperature rise calculated by the entering temperature-temperature rise curves, and calculated by the temperature-length curves was 6.5%. The duct air temperature measurements were accurate within ±0.3 F so that an error of 0.6 F could occur, which is only

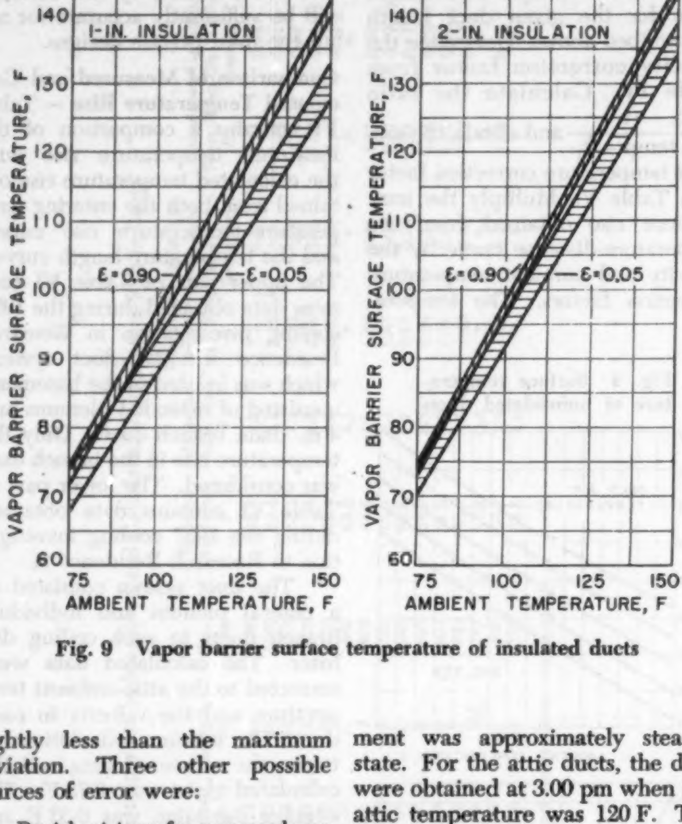


Fig. 9 Vapor barrier surface temperature of insulated ducts

slightly less than the maximum deviation. Three other possible sources of error were:

1. **Duct heat transfer measured was not absolutely steady-state.**
2. **Basement and attic air temperatures were not necessarily uniform throughout the spaces.**
3. **Mean radiant temperature was not necessarily equal to the ambient air temperature**

The basement air temperature was relatively constant and the compressor had been operating several hours prior to the period of the study so that the heat transfer to the ducts located in the basement was approximately steady-state. For the attic ducts, the data were obtained at 3.00 pm when the attic temperature was 120 F. The maximum attic air temperature, 123 F, occurred at 1:45 pm so that the heat transfer rate was influenced to some extent by the maximum temperature. The average surface temperature, which is an index of the mean radiant temperature, was 122 F when the attic air temperature was 120 F. The calculated temperature rise data were sufficiently accurate for design purposes.

Condensation on Duct Surfaces — The data presented in the preceding sections are based upon the assumption that condensation does not occur on the duct or insulation surfaces. This will be true if the exposed surface temperature is above the dew point of the ambient air. Figs. 8 and 9 show the duct surface temperature for uninsulated ducts and vapor barrier surface temperature for insulated ducts. The surface temperatures are independent of duct diameter.

For uninsulated ducts, the surface temperature is a function of the duct air temperature, ambient temperature and duct air velocity. The surface temperatures of uninsulated ducts exposed to ambient temperature from 75 to 150 F are shown in the upper part of Fig. 8 for a duct air velocity of 800 fpm with duct air temperature as a parameter. The surface temperatures range from 56 to 94 F. For duct air velocities other than 800 fpm the surface temperature must be adjusted by the correction indicated in the lower part of Fig. 8. The surface temperature correction varies from 10.6 to −5.7 F, depending upon the velocity and ambient temperature.

The vapor barrier surface temperatures for insulated ducts are shown in Fig. 9. The surface temperature was practically independent of duct air temperature and velocity within the range investigated. It was principally dependent upon ambient temperature, surface emissivity and insulation thickness. The surface temperatures are shown as a function of ambient temperature with bands for each vapor barrier emissivity encompassing all variations due to duct air temperature and velocity. The lower emissivity, in conjunction with reducing the temperature rise in ducts, causes a lower surface temperature. The lower part of the bands are for lower duct air temperature and higher velocities. As stated earlier, condensation will occur if the exposed surface temperature is equal to or less than the dew point of the ambient air.

ACKNOWLEDGMENTS

This paper reports one phase of a comprehensive investigation conducted in Warm Air Heating Research Residence No. 2 under the terms of a cooperative agreement between the National Warm Air Heating and Air Conditioning Association and the Engineering Experiment Station of the University of Illinois. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

Acknowledgment is made to the manufacturers who cooperated by furnishing equipment used in the investigation.

REFERENCES

1. Gilkey, H. T., D. R. Bahnfleth, and R. W. Roose, "Cooling a Small Residence with a Two-Horsepower Mechanical Condensing Unit." ASHVE TRANSACTIONS, Vol. 59, New York, 1953.
2. Kratz, A. P., S. Konzo, and R. B. Engdahl, "Temperature Drop in Ducts for Forced-Air Heating Systems." University of Illinois Engineering Experiment Station Bulletin 351, 1944.
3. McAdams, William H., Heat Transmission, McGraw-Hill Book Company, Inc., New York, Third Edition, 1954.
4. Keenan, Joseph H. and Joseph Kaye, Gas Tables, John Wiley and Sons, Inc., New York, 1948.

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y.: In office buildings and to a lesser extent in homes, where there is a series of branch take-offs from the trunk duct, it was observed a number of years ago that the temperature change down the duct was far less than calculated. This was investigated, both theoretically and experimentally, and it was indicated that the reason for this is that most of the heated or cooled air is accumulated along the outside of the duct; therefore, when air is taken from the outside of the duct, depending upon the character of the branch take-off, a substantial amount of the heating or cooling effect which was added to or subtracted from the duct is removed. Under idealized conditions, the take-off temperature can be maintained almost constant throughout the length of a duct of considerable length. For instance, with 10-ft intervals between take-offs for a 100-ft whole length of duct, it is possible that except for some minor deviations, even with an uninsulated duct, a virtually constant take-off temperature can be maintained.

R. LANDSBERG, Haifa, Israel (Written): I have dealt with a similar problem and the results for uninsulated ducts without condensation are extremely close to those obtained in this study, but it would appear that the influence of condensation should be considered as well.

As an example, an uninsulated square duct of 15 x 15 ft with 90 F outside and 55 F inside at 800 fpm air velocity, would have a surface temperature of 64 F. This is the dew point for 42% relative humidity, but higher humidity is by no means improbable. Thus, condensation would occur and affect the results as follows:

Outside humidity %	42	50	60	90
Surface temperature F	64	66	58	73
Total heat transfer (Btu/sq ft/hr)	22	24	30	40
Moisture condensed (lb/sq ft/hr)	0	0.007	0.013	0.026

The numerical results are less important than the obvious conclusion that uninsulated ducts can be used only when the risk of condensation is excluded.

AUTHOR WRIGHT: It is certainly true that condensation will probably occur on uninsulated ducts located in ambient conditions which approach outdoor conditions. This is one of the reasons that it is always recommended that ducts which pass through non-conditioned spaces be insulated.

Condensation should also be prevented because of the effect of the water on construction materials. The problem of condensation occurring was not considered in this paper because this is not recommended practice in most air conditioning applications.

1764

No. 1764

A Unique Hot-Box Cold-Room Facility

W. P. BROWN

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Member ASHRAE

A. G. WILSON
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A unique guarded hot box has been designed and built recently by the Division of Building Research of the National Research Council to be used in conjunction with a large cold room (Fig. 1) for the measurement of the heat transmission coefficients of building sections 4 ft wide and 8 ft high. A section to be tested is incorporated as part of a partition which separates the cold room into two compartments. The smaller compartment, or warm room, then is heated electrically to maintain any desired constant warm side temperature from 65 to 75 F. The larger compartment is refrigerated to maintain the desired constant cold side temperature from 40 to −60 F.

The guarded hot box (Figs. 2 and 3) is positioned against the warm side of the building section and is heated electrically to maintain a constant temperature. The heat transmission coefficient is calculated from the measured electrical input and the temperatures.

The guarded hot box is similar in design to the warm box of the large-scale wall heat-flow measuring apparatus described in a previous paper.[1] The design overcomes some of the limitations of other methods of determining heat transmission coefficients.

The test area of the box is large enough to meter the heat flow into the whole of representative sections of building walls. Also building sections much wider than 4 ft can be installed in the cold room partition and heat flows measured at different positions. In this way the effect upon heat transmission coefficients of non-uniformities in construction can be studied.

Building sections tested are exposed on both sides to conditions closely simulating those used for design calculations. The section is exposed to controlled temperature surroundings as well as to air at controlled temperature.

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This paper was prepared for presentation at the ASHRAE 68th Annual Meeting, Denver, Colo, June 26-28, 1961.

Air motion on the cold side is rapid and the outside surface conductance is realistic. The box is large enough so that the surface conductance at the warm side of the test wall under natural convection conditions approaches that occurring in practice.

The apparatus has been used successfully for the measurement of heat transmission coefficients of several metal skinned curtain walls. Additional tests on uniformly constructed walls were made to assess the performance of the apparatus. Studies were made of the inside surface conductance and of errors due to heat leakage through the box walls and at the edges. The apparatus and its operation now are described, together with results of the tests.

Description of cold room—Cooling in the cold room is provided by a diffuser containing the evaporator of the refrigeration system, past which air is circulated by a blower. The air is discharged horizontally near the ceiling through the supply grille. The grille has vertical adjustable vanes which distribute the air evenly across the room. The air returns through an opening in the front of the diffuser near the floor.

In operating the cold room under given conditions, the refrigeration system is run continuously to provide an almost constant cooling effect, which is in excess of that required. Temperature control is achieved by reheating the air by electric heaters after it passes the evaporator. A resistance-element-type recorder and a three-action (proportional + reset + rate) controller regulate the amount of reheat. The air temperature varia-

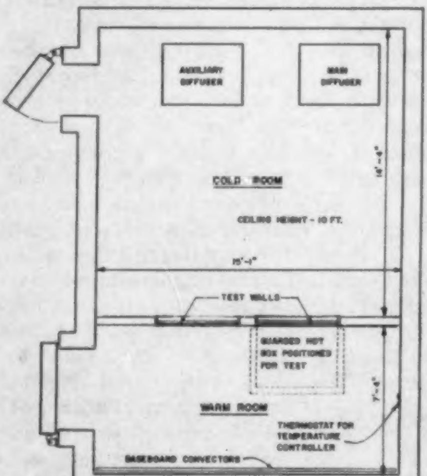


Fig. 1 Plan view of cold room

tion at any point in the cold room is less than ±0.1 F. Floor-to-ceiling temperature gradients are less than 0.5 F. Air temperature can be controlled anywhere from −60 to 40 F on the cold side of a test wall.

For guarded hot box tests the warm room is heated by electric gravity baseboard convectors located along the wall opposite the test partition. The input voltage to the convector is regulated by a temperature controller. At the thermostat, air temperatures are controlled to ±0.25 F. Air temperature gradients depend upon air circulation.

Description of guarded hot box apparatus – This apparatus is a large box mounted on wheels which can be rolled against the warm side of a test wall installed in the cold room partition. The measuring area of the box is 4 ft wide and 8 ft high. The depth of the box is 3 ft.

The inside of the box is lined with aluminum panels (Fig. 4). The panels, which were developed for radiant heating and cooling applications, consist of aluminum extrusions with holders for copper tubing, through which water is circulated. A similar set of panels are installed outside of the inner panels and are separated from them by thermal insulation. When the outer panel is maintained at the same temperature as the inner panel, a guard is formed which prevents heat transfer across the walls of the box. The outside of the guard panel is insulated and sheathed with plywood. The contact edge of the box is ¾ in. wide and is formed by wood blocking between the inner panel and an extension of the guard panel.

A water reservoir, pump, and pump motor for circulating water through the tubes of the inner panel are located inside the box. The pump is belt-driven by a series-wound dc motor. The energy input to the motor can be varied from 25 watt (85 Btu/hr) to 350

Fig. 2 View of guarded hot box in front of cold room

watt (1200 Btu/hr) by regulating the voltage to the motor from 15 to 70 volt. The electrical input to the pump is normally sufficient to provide all of the wall transmission; thus the energy imparted to the water by the pump serves to heat the panel and the energy loss from the motor and drive serves to heat the air directly.

Air heaters and water heaters are provided to supply an additional 150 watt (500 Btu/hr) of energy and to vary the ratio of air heat to panel heat if so desired. The motor, pump, and air heaters which operate at higher than panel temperature, are surrounded by an additional set of panels through which water is circulated to prevent direct radiant exchange with the test wall surface.

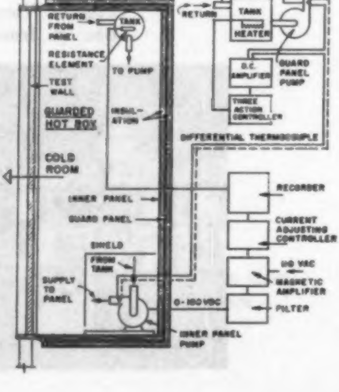
A second reservoir and pump, located outside the box in the warm room, supply liquid (2/3 glycol + 1/3 water) to the guard panel. The liquid is continuously cooled by circulating part of it through a section of finned tubing mounted on the ceiling of the cold side. A hand valve regulates the flow of liquid through the tubing. The reservoir contains an electric heater to control the temperature of the liquid.

Instrumentation and control – The control system for the guarded hot box is shown in block form in Fig. 3. The dc input to the inner panel pump motor and heaters is supplied by a 500-watt (100 volt and 5 amp) magnetic amplifier, the output power of which is regulated by a resistance-element-type recorder and current adjusting controller. The controller produces an output signal of 0 to 5 milliamp and by manual switching all or part of the signal is applied to the control winding of the magnetic amplifier to regulate its output voltage from approximately 15 to 100 volt or 15 to 70 volt. This, together with resistance heaters which can be connected in series with the pump motor, permits regulation of the maximum current to 5 amp and power regulation from 25 to 500 watt or 25 to 350 watt.

The recorder has a 10 F span (65 to 75 F) and a sensitivity of 0.01 F or better. Operating records have shown that the inner panel temperature is controlled to within 0.01 F of set value.

The input voltage and current to the inner panel are measured by a millivolt recorder connected across appropriate resistors. The

Fig. 3 Schematic cross section of apparatus

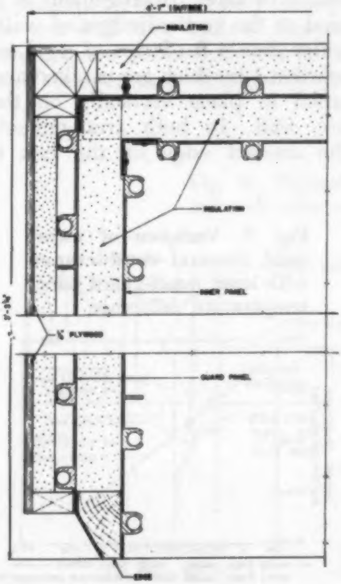


maximum recorder error, with calibration, is limited to less than ± 0.25 per cent of scale span so that by sizing the resistors to give larger than half-scale deflection this error is always less than 0.5 per cent.

A filter network on the output of the magnetic amplifier limits ripple voltage to less than 0.75 volt. It can be shown that the error due to the ac component is negligible for all power inputs. The total error in power measurement is less than 1.0 per cent.

A servo voltage multiplier is being procured so that input power can be recorded directly. This will improve the accuracy of measurement and will facilitate the determination of average power input.

Fig. 4 Details of guarded hot box walls and edge



A differential thermocouple signal is amplified and used to actuate a three-action controller to control guard panel temperature. The controller regulates an electric heater. Heating can be supplied continuously on, continuously off or pulsed on-off by the controller. The sensitivity of the amplifier-controller arrangement is such that guard panel liquid temperature is controlled to within ± 0.05 F or less of inner panel liquid temperature at the respective panel headers.

The rates of circulation are such that the temperature drops through the two panels are very small, less than 0.1 F. Also, the panel to water temperature difference and the temperature variation over the panel surface is less than 0.1 F. It is unlikely, therefore, that the average temperature difference between the inner and guard panels exceeds 0.05 F.

Temperature measurements are made with 30-gauge copper-constantan thermocouples and an electronic self-balancing temperature indicator. With calibration of the apparatus, temperature difference measurements are accurate to ± 0.2 F. This corresponds to an error of ± 0.67 per cent at an overall temperature difference of 30 F.

Heat leakage – There are two sources of heat leakage in the guarded hot box. Panel heat leakage, leakage of heat between the guard and inner panels, is caused

by temperature differences between the two panels. Edge heat leakage is lateral heat flow at the junction of the hot box edge construction and test wall which results in an error in metered heat into the test area.

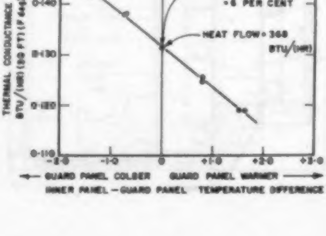
Panel heat leakage—The insulation between the inner and guard panels provides a calculated thermal conductance of 0.18 Btu/hr/sq ft/(F temperature difference) or a total heat flow of 19 Btu/hr/(F temperature difference). Tests were carried out on a wall of uniform thermal conductance to determine the actual panel heat leakage for a number of guard panel/inner panel temperature differences. The results are shown on Fig. 5. The variation in thermal conductance of 6 per cent per F temperature difference for the test wall denotes a panel heat leakage of 22 Btu/hr/(F temperature difference). With the panel temperatures balanced to ± 0.05 F the actual panel heat leakage is ± 1.1 Btu/hr as compared with the estimated heat leakage of ± 0.9 Btu/hr.

Edge heat leakage—The partition framework in the cold room is constructed to accommodate test walls 8 ft high with an allowance of 1 in. for clearance at the top and the bottom. Movable vertical posts can be positioned for various test wall widths.

It is possible to install walls from 4 to 12 ft in width. Which width is preferable depends upon the thermal characteristics of the test wall and upon practical considerations of wall fabrication. With walls having modules other than 12, 16 or 24 in. for example, it may be desirable to have a test wall wider than 4 ft and to make tests at more than one position, in order to determine the effect of joints or other discontinuities on the thermal coefficients.

Fig. 6 shows the two edge arrangements that have been used for guarded hot box tests to date. Arrangement A is used at the top and bottom of all test walls and at the sides of test walls which are 4 ft wide. The test wall is separated from the surrounding partition by a 1-in. thickness of rigid insulation. The wall is held in place in the partition by compression. All possible sources of mass transfer into or out of the box are sealed with either caulking compound or tape. Arrangement B is used at the vertical edges of walls wider than 4 ft. Strips of rigid insulation 1 in. thick are caulked and taped in place vertically on the test wall. In both arrangements the contact edge of the box is

Fig. 5 Variation of measured thermal conductance with inner panel-guard panel temperature difference



clamped against the protruding edge of the insulation. The box is sealed to the edge of the insulation with tape.

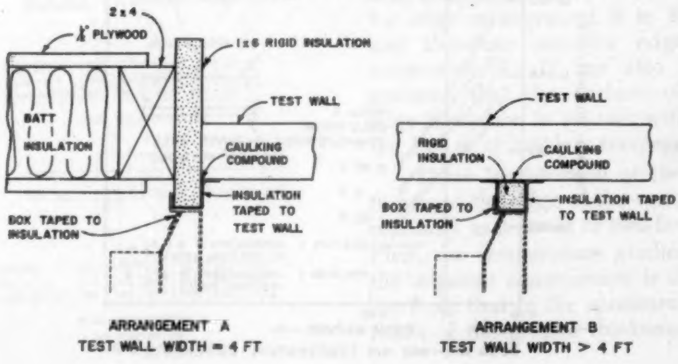
Whichever edge arrangement is used, the edge heat leakage error is a function of the average air temperature difference across the edge. In Fig. 7 measured thermal conductance is plotted against warm room air/guarded hot box air temperature difference for several test walls. The box air temperature is the average air temperature 3 in. from the test wall surface and the warm room air temperature is the average air temperature surrounding the guarded hot box edge, measured with thermocouples.

There is a vertical air temperature gradient in the hot box dependent on the heat flow through the test wall. For walls of thermal conductance about 0.15 Btu/hr/sq ft/(F temperature difference), the air temperature at the top edge is about 2.5 F higher than that at the bottom edge, with a warm side/cold side temperature difference of 70 F. The air temperature gradient in the warm room around the edge of the guarded hot box is similar to that in the hot box.

This is achieved despite the confining effect of the hot box edge construction by providing gentle forced circulation with a propellor fan, which is directed to supplement the natural air motion in the warm room induced by the baseboard convector on the wall opposite the test specimen.

All the test walls show a marked variation of measured thermal conductance with air temperature difference between hot box and warm room. The degree of variation depends on test wall construction and on the type of edge arrangement that is used. Walls 1 and 2 of Fig. 7 are almost

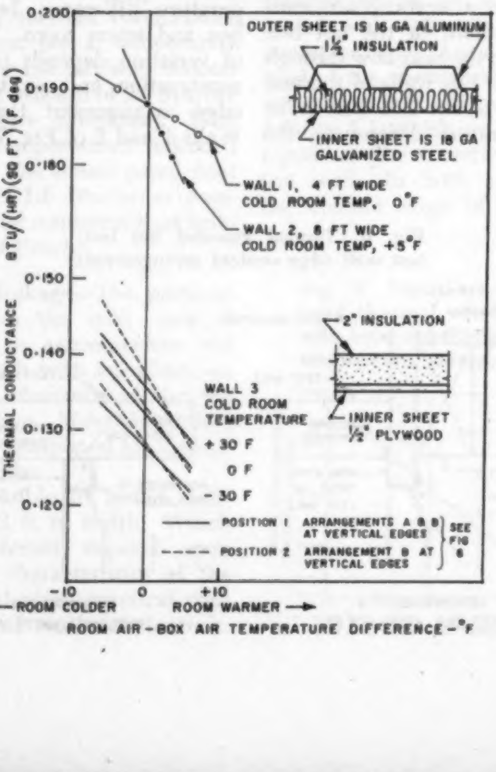
Fig. 6 Typical guarded hot box-test wall edge contact arrangements



identical curtain walls. Wall 1 was 4 ft wide and edge arrangement A was used at the vertical edges. Wall 2 was 8 ft wide and edge arrangement B was used at the vertical edges. The metal skin provided a highly conductive heat transfer path across edge arrangement B. Consequently, the lateral heat transfer or edge heat leakage across edge arrangement B was greater than the lateral heat transfer across edge arrangement A for a given air temperature difference.

Wall 3 of Fig. 7 is an 8- by 8-ft test wall of uniform thermal conductance. At position 1, edge arrangement A was used at one vertical edge and edge arrangement B at the other. At position 2, edge arrangement B was used at both vertical edges. The edge heat leakage at position 2 was greater than the edge heat leakage at position 1 for the same air temperature difference, since the resistance to lateral heat flow through the plywood surface in edge arrangement B was less than through the perimeter insulation in edge

Fig. 7 Variation of measured thermal conductance with room air-box air temperature difference

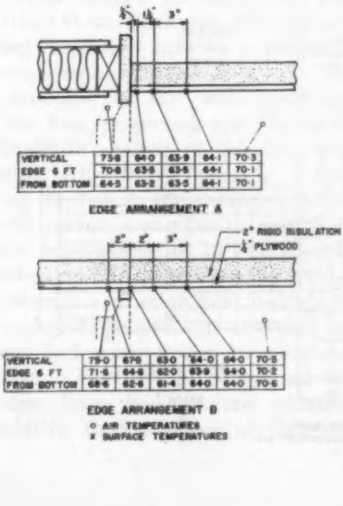


arrangement A.

With the warm room air temperature substantially higher than the box air temperature, the edge heat leakage results in a decrease in metered heat flow into the test area and vice versa. It would be desirable if there were no lateral heat flow at the edges of the test section with hot box and warm room at the same temperature. The surface temperature pattern at the edge would then be unaffected by the presence of the box and conditions would be ideal for measuring a correct value of thermal conductance.

Fig. 8 shows surface temperatures measured at the vertical edges of wall 3 with different warm room air temperatures for both edge arrangements A and B. At small values of hot box/warm room air temperature difference (up to 5 F)

Fig. 8 Edge surface temperatures



the surface temperature 5 in. from the edge was unaffected by the edge heat exchange. The distortion of the surface temperature pattern from this point to the edge at equal warm room and hot box temperatures was quite small, except at the junction of the specimen and perimeter insulation; edge heat leakage resulted in an increase in heat flow at the inner surface in both edge arrangements. Similar measurements indicated a comparable increase in heat flow at the top edge and perhaps a small decrease in heat flow at the bottom edge with equal average warm room and hot box air temperatures.

To assess the error due to edge heat leakage a relaxation calculation[2] was carried out for a homogeneous wall with edge arrangement A. The section studied is shown in Fig. 9 along with the temperature distribution obtained. The calculation has been made for equal air temperature in warm room and hot box and equal surface conductances. This latter assumption can be shown to be reasonably valid for the air and surface temperature conditions shown for edge arrangement B in Fig. 8 and therefore also for edge arrangement A. It has also been assumed that the surface of the edge insulation in contact with the hot box is at hot box temperature.

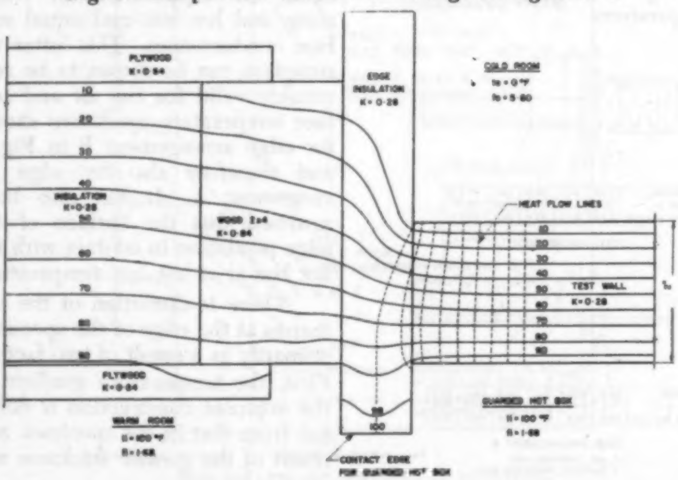
There is distortion of the isotherms at the edge of the specimen primarily as a result of two factors. First, the temperature gradient in the adjacent construction is different from that in the specimen as a result of the greater thickness and

thermal resistance of the former. Consequently, all points through the depth of the specimen tend to be at a slightly higher temperature at the edge than at similar locations removed from the edge with a resulting tendency for lateral heat flow into the test section. Counteracting this is the effect of the projection of the edge insulation on the warm side which tends to lower temperatures at the adjacent inner surface of the specimen. The net effect is a small increase in metered heat flow from the hot box into the test area of the specimen.

The increased heat flow at the edge of the specimen for arrangement A, as determined from the assumed inside surface conductance and the calculated surface temperatures in Fig. 9 represents 0.3 per cent of the total heat flow into the test area. On the other hand, the heat flow from the hot box into the edge insulation is 2.7 per cent of the total heat flow into the test area. The calculation thus indicates that with the hot box and the warm room air temperatures balanced, the error in measured thermal conductance due to edge heat leakage is +3 per cent. This is confirmed by the calculated value of thermal conductance for the test wall, based on measured values of conductivity for the insulation and published values of thermal conductance for plywood, which is 3 to 4 per cent lower than the value obtained in the guarded hot box.

No relaxation calculation has been made of the edge heat leakage error for edge arrangement B. The isotherm pattern, which can be visualized by reference to that

Fig. 9 Temperature distribution at edge arrangement A



for arrangement A in Fig. 9, will be affected only by the 1-in. square strip of edge insulation and will be symmetrical about it. The distortion of the isotherms at the inner surface of the specimen adjacent to the edge will be similar to that in Fig. 9 but slightly more pronounced, since there will not be the counteracting effect of the thicker edge construction. Thus, for thin walls at least, the error due to edge heat leakage with warm room and hot box air temperatures balanced is likely to be slightly greater. This is confirmed by the results in Fig. 7, which for wall 3 shows slightly higher values of thermal conductance at balance with arrangement B at both vertical edges.

The error due to edge heat leakage can be minimized by reducing the projection of the edge insulation from the inner surface of the test specimen. Ideally, this should approximate the inside surface conductance of the specimen, which would be achieved with about ¼-in. thickness. The projecting insulation acts as a gasket to overcome irregularities in the planeness of the wall specimen. The 1-in. projection was chosen to facilitate sealing of the box with tape. Alternative methods of sealing the box to the specimen which will permit a smaller projection of the edge insulation are being considered. This is expected to reduce errors due to edge heat leakage to negligible proportions at equal hot box and warm room air temperatures. Further investigations of edge heat leakage are planned relative to the effect of different surface conductances and air temperatures with various wall constructions and edge arrangements. It is thought that the resistance paper analog might be adapted for this study. It will be recognized that edge heat leakage is equally a problem with the traditional guarded hot box.[3]

WARM SURFACE HEAT EXCHANGE

Heat transfer characteristics of wall constructions usually are specified or compared on the basis of the thermal transmittance coefficients or U values, which include the surface conductances. Since the surface conductance is a function of surrounding conditions, it is necessary to use standard values. In the ASHRAE GUIDE inside surface conductances are based on natural convection, with air and surroundings at the same temperature. In the traditional guarded hot box, forced convection is provided, so that the inside surface conductances are usually much higher than those for natural convection. It is necessary, therefore, to base the thermal conductance of the wall on measured surface temperatures and to compute the U value, incorporating the standard inside surface conductance.

The inside surface temperature distribution and therefore the thermal conductance of non-uniform walls is affected by the inside surface conductance. Thus the thermal conductance obtained with forced convection on the inside may differ somewhat from that for natural convection. Furthermore, it is sometimes difficult to establish an appropriate average surface-to-surface temperature difference by direct measurement for the determination of thermal conductance of non-uniform walls because of the variations in surface temperature.

It is therefore desirable in heat flow measurements to provide surface conductances closely approximating standard values used for calculating U values and to know the surface heat exchange characteristics of the apparatus precisely, in order to facilitate the determination of thermal conductance.

The total heat exchange between a test wall surface and the guarded hot box is the sum of the convection exchange between the surface and the box air and the radiation exchange between the surface and the box inner panel surface.

$$q_t = q_r + q_c \tag{1}$$

In a grey enclosure, such as the guarded hot box, the radiation exchange[4] is:

$$q_r = F_{wp}\,\sigma\,(T_p^4 - T_w^4) \tag{2}$$

For one source (guarded hot box) and one sink surface (test wall) and no interconnecting surfaces, it can be shown that

$$F_{wp} = \frac{1}{\dfrac{1}{e_w} + \dfrac{A_w}{A_p}\left(\dfrac{1}{e_p} - 1\right)} \tag{3}$$

The radiation component can also be accounted for by the following:

$$q_r = h_r\,(t_p - t_w) \tag{4}$$

where

$$h_r = F_{wp}\,\sigma\,\frac{(T_p^4 - T_w^4)}{(T_p - T_w)} \tag{5}$$

Equation (5) can be approximated by the following:

$$h_r = 4\,F_{wp}\,\sigma\,(T_m)^3 \tag{6}$$

For the range of conditions in the guarded hot box tests, the error in using equation (6) is less than 0.05 per cent.

The heat transfer between the air in the guarded hot box and test wall surface is by natural convection. The convective movements are brought about solely by differences of density caused by differences of temperature. Natural convection can be expressed in dimensionless terms by relating the Nusselt, Grashof and Prandtl numbers as follows[5]

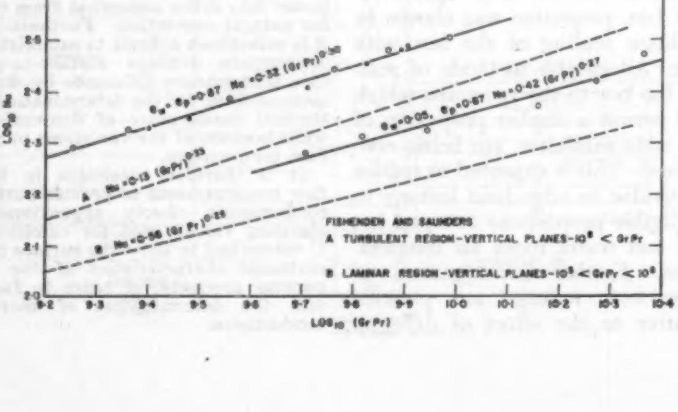
$$Nu = x\,(Gr\,Pr)^y$$

$$\frac{h_c L}{k} = x\left[\left(\frac{C_p\,\mu}{k}\right)\left(\frac{\beta\,\rho^2\,g L^3\,(t_a - t_w)}{\mu^2}\right)\right]^y \tag{7}$$

The constants x and y are evaluated from experimental data. The equation can be applied to all tests in the guarded hot box provided that the geometry of the system remains the same and (Gr Pr) lies within the range of conditions used in the original evaluation of the constants.

The bulk temperature of the fluid, well away from the surface, is used in the determination of the temperature difference $(t_a - t_w)$. The values of the physical constants C_p, ρ, k, and μ

Fig. 10 Heat exchange in guarded hot box by natural convection



are taken at the arithmetic mean temperature of surface and bulk fluid. The height of the wall surface is taken as the characteristic linear dimension, L.

Because the temperature range in the guarded hot box tests is small, the physical properties of air may be regarded as constants. Equation (7) can therefore be simplified to:

$$h_c = x' (t_a - t_w)^y \tag{8}$$

where

$$x' = \frac{xk}{L} \left[\left(\frac{C_p \mu}{k} \right) \left(\frac{\beta g \rho^2 L^3}{\mu^2} \right) \right]^y \tag{9}$$

The heat flow rate due to convection is:

$$q_c = h_c (t_a - t_w) \tag{10}$$

The total heat flow rate from equations (1), (4) and (10) is

$$q_t = h_r (t_p - t_w) + h_c (t_a - t_w) \tag{11}$$

Combining h_r and h_c to form the usual surface conductance f_i and defining an equivalent temperature t_i, equation (11) becomes:

$$q_t = f_i (t_i - t_w) \tag{12}$$

The equivalent temperature, t_i, is the temperature of air and panel surface which would produce the same total heat flow rate as t_p and t_a and is calculated from equations (11) and (12)[1].

$$t_i = \frac{h_r t_p + h_c t_a}{h_r + h_c} \tag{13}$$

For a test on a wall of uniform conductance where it is possible to measure the average surface temperature, t_w, accurately, q_r can be readily calculated from equation (2), knowing the emissivities and interchange factor. Since q_t is accurately measured, q_c and hence h_c can be determined from equations (1) and (10). A number of values of h_c can be obtained from tests at a number of different total heat flow rates. The dimensionless groups from equation (7) can then be evaluated for each of the tests and the constants x and y determined by correlation.

The following three series of tests were carried out with the guarded hot box to evaluate the convection constants:

1. Tests at different total heat flow rates on a uniform 4- by 8-ft wall painted flat white ($e_p = e_w = 0.87$)
2. Tests at different total heat flow rates on same wall as above but covered with aluminum foil ($e_p = 0.87$, $e_w = 0.05$)
3. Tests at different total heat flow rates on a uniform 8- by 8-ft wall painted flat white ($e_p = e_w = 0.87$)

The total hemispherical emissivities quoted were estimated from the measured[6] total normal emissivities. For this purpose the total hemispherical emissivity was taken as 0.95 and 1.15 of the total normal emissivity for nonconductive and conductive surfaces, respectively[7]. The over-all interchange factors were calculated from equation (3). For all tests $A_w = 32$ sq ft, $A_p = 137$ sq ft for series 1 and 2 and 153 sq ft for series 3. The change in box surface area was due to extensive modifications carried out between series 2 and 3 tests. Average air and surface temperatures were measured with thermocouples. The temperature of the inner panel circulating fluid was taken as the average temperature of the inner surface of the guarded hot box. All air properties were taken from reference 8.

The test results are shown plotted in Fig. 10. Curves from Fishenden and Saunders also are given. There is substantial disagreement between the results for the high and low emissivity surfaces. For this reason the results for each were correlated separately by the method of least squares. The analysis indicated that convection to the test wall surfaces may be represented by the following equations:

For $e_w = e_p = 0.87$

$$Nu = 0.32 \, (Gr\,Pr)^{0.28} \tag{14}$$

For $e_w = 0.05$, $e_p = 0.87$

$$Nu = 0.42 \, (Gr\,Pr)^{0.27} \tag{15}$$

Considering air properties as being constant the equations can be simplified to the form of equation (8) as follows:

For $e_w = e_p = 0.87$

$$h_c = 0.29 \, (t_a - t_w)^{0.28} \tag{16}$$

For $e_w = 0.05$, $e_p = 0.87$

$$h_c = 0.21 \, (t_a - t_w)^{0.27} \tag{17}$$

The difference in the rate of convection heat exchange for the high and low emissivity surfaces can be explained by reference to the various heat exchanges taking place in the guarded hot box, shown in Fig. 11. A summation of the heat exchanges gives:

$$q_t = q_r + q_c = q_r + q'_c + q_L \qquad (18)$$

hence

$$q_c = q'_c + q_L \qquad (19)$$

Test results show that the panel surface to air temperature difference $(t_p - t_a)$ is quite small. Furthermore, at equal air to wall surface temperature difference $(t_a - t_w)$, the difference between $(t_p - t_a)$ for the high and low emissivity wall surface is negligible. It can be seen from equation (19), therefore, that the difference in the convection exchange, q_c, at equal values of (Gr Pr) for the high and low emissivity wall surfaces is due to differences in the motor loss q_L. Since the motor loss increases with increasing total heat input, q_c increases proportionately. The increased radiation exchange to the high emissivity surface requires a larger total heat input. The convection heat rate, q_c, is therefore larger for the high emissivity surface.

The equation to be used for the calculation of the convective component of the surface conductance therefore depends upon test wall emissivity. The two equations determined here correspond to the extremes of surface emissivity likely to be encountered in guarded hot box tests.

The convection equations compare favorably with the recommended equation for turbulent flow on vertical free plates (curve A Fig. 10). This is expected since turbulence is known to set in at (Gr Pr) about 10^9 although at values of 10^9 to 10^{10} it is still fully developed only at the upper parts of the surface. It is thought that the over-all effect of the boundaries is to reduce the convection heat transfer below that expected on a free plate of the same dimensions. This is offset, however, by the presence of the pump motor, which increases the convection heat transfer by increasing circulation.

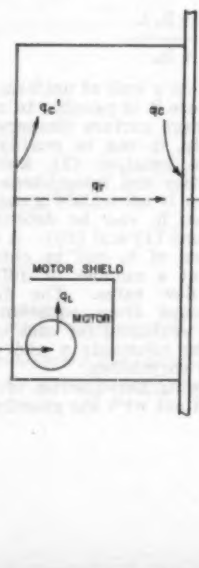
Equations (6), (16) and (17) are used in calculating surface conductances for guarded hot box tests. Where the wall surface emissivity falls between values for which the convection equations are valid, the surface can usually be painted to provide an emissivity approximately equal to 0.87. These equations can also be used to determine an average surface temperature for non-uniform walls from which a wall thermal conductance can be calculated. Rearranging equation (11) gives

$$t_w = \frac{h_r t_p + h_c t_a - q_t}{h_r + h_c} \qquad (20)$$

from which t_w can be obtained by trial and error.

The surface conductances provided in the hot box tests compare favorably with those recommended by the ASHRAE GUIDE[9] for purposes of computation. For a surface of emissivity 0.90 and temperature 70 F facing surroundings having an emissivity of 0.90 and at the same temperature as the ambient air, the GUIDE recommends a combined surface conductance of 1.46 based on a surface-air temperature difference of 10 F. For the same conditions the surface conductance with the guarded hot box is 1.45.

Fig. 11 Heat exchange in guarded hot box



COLD SURFACE HEAT EXCHANGE

Forced convection is provided in the cold room by air movement induced by the diffuser fans. Each fan motor has three speed settings and the total air discharge rate can be varied from 2800 to 11,400 cfm in six steps. The discharge rate decreases slightly with frost build-up on the evaporator coils. Tests on smooth-surface uniform walls with emissivity equal to 0.9 have shown that outside surface conductance, f_o, varies between 3 and 4 with the main diffuser fan operating at high speed.

The variation in surface conductance with heat flow, temperature level and wall emissivity is rather small since the heat transfer is mainly by forced convection. Calculations for a typical wall show that the radiant component of the surface heat exchange may vary from 0.4 to 0.75 for a wall of 0.9 emissivity and from 0.1 to 0.2 for a wall of 0.25 emissivity as the cold room temperature varies from —60 to 40 F.

CONCLUSION

It has been shown that heat flow through any 4- by 8-ft section of test walls up to 12 ft in length can be metered reliably with the guarded hot-box cold-room arrangement described in this paper. The total error in determination of thermal transmittance coefficients depends upon test conditions and on wall thermal conductance. For a wall of U value 0.1 Btu/hr/sq ft/F at a temperature difference of 30 F the total of panel heat leakage, control and measurement errors, if additive, is ± 2.8 per cent. The edge heat leakage error depends upon wall construction and edge arrangement. For thin wall sections with the edge configuration used to date it is about $+3$ per cent.

The error in measured thermal transmittance coefficient for such walls is therefore between $+0.2$ per cent and $+5.8$ per cent. Alternative methods of sealing the box to the wall specimen which will largely eliminate edge heat leakage error are being considered. The total percentage error will decrease as the temperature difference between hot box and cold room increases or as the thermal conductance of the wall increases.

The error in measuring thermal conductance depends on the reliability with which an appropriate average surface temperature can be measured or estimated. Operation indicates that heat transmission coefficients can be reproduced within 1.0 per cent.

The temperature of the inside surface of the hot box is under direct control. This makes it possible to operate the box under conditions of natural convection without excessive vertical air temperature gradients and to exercise some control over the relationship between box surface and air temperature. With energy supplied only to the pump motor the air temperature has been generally within ½ F of the box surface temperature in tests on typical walls.

Relationships describing the radiant and convective components of the inside surface conductance, applicable to most walls, have been determined for the box. These relationships can also be used to determine an average inside surface temperature for non-uniform walls from which a wall thermal conductance can be calculated. The inside surface conductances provided by the box are similar to

those that occur in practice under natural convection conditions. Realistic outside surface conductances can be provided in the cold room.

ACKNOWLEDGMENTS

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NOMENCLATURE

A = area of a surface, sq ft
- A_p, inside surface of guarded hot box
- A_w, surface of test wall

C = thermal conductance, Btu per (hr) (sq ft) (F temp diff)

C_p = specific heat of air at constant pressure, Btu per (lb) (F)

e = total hemispherical emissivity of a surface, dimensionless e_p, inside surface of guarded hot box
- e_w, surface of test wall

f = combined surface film conductance, Btu per (hr) (sq ft) (F temp diff)
- f_i, warm surface; f_o, cold surface

F_{wp} = over-all interchange factor for radiation from test wall to guarded hot box, dimensionless

g = local acceleration of gravity, ft per (sec²)

Gr = Grashof number, $\beta\, g\, \rho^2 L^3 (t_a - t_w)/\mu^2$, dimensionless

h = surface conductance, Btu per (hr) (sq ft) (F temp diff)
- h_c, convection; h_r, radiation

k = thermal conductivity of air, Btu per (hr) (sq ft) (F per ft of thickness)

K = thermal conductivity, Btu per (hr) (sq ft) (F per in. of thickness)

L = characteristic linear dimension for convection, ft

Nu = Nusselt number, $h_c L/k$, dimensionless

Pr = Prandtl number, $\mu C_p/k$, dimensionless

q = heat flow rate, Btu per (hr) (sq ft)
- q_c, convection heat exchange between box air and test wall surface
- q'_c, convection heat exchange between box inner panel surface and box air
- q_L, convection heat exchange between box motor and box air
- q_r, radiation heat exchange between box inner panel surface and test wall surface
- q_t, total heat exchange between box and test wall, also equal to total heat input to box

T = absolute temperature, R (deg)
- T_a, average air temperature in guarded hot box
- T_p, average temperature of inner panel surface of guarded hot box
- T_w, average temperature of test wall surface
- T_m, mean temperature between inner panel surface and test wall surface

t = temperature, F (deg)
- t_a, average air temperature in guarded hot box
- t_i, temperature equivalent to t_p and t_a
- t_o, average air temperature in cold room
- t_p, average temperature of inner panel surface of guarded hot box
- t_w, average temperature of test wall surface

U = thermal transmittance, Btu per (hr) (sq ft) (F temp diff)

x, x′, y = constants

Greek Letters

β = coefficient of thermal expansion of air, $1/T_a$ per R

σ = Stefan-Boltzmann constant 0.1712×10^{-8} Btu per (hr) (sq ft) $(R)^4$

μ = dynamic viscosity of air, lb per (hr) (ft)

ρ = mass density of air, slugs per $(ft)^3$

BIBLIOGRAPHY

[1] Large-Scale Wall Heat-Flow Measuring Apparatus, K. R. Solvason. (ASHRAE Transactions, Vol. 65, 1959, p. 541-550).

[2] Relaxation Methods, D. Allan. (McGraw-Hill, New York, 1954).

[3] Method of Test for Thermal Conductance and Transmittance of Built-up Sections by Means of the Guarded Hot Box. ASTM Designation C236-54T.

[4] Heat Transmission, W. H. McAdams. (McGraw-Hill, New York, 1954, p. 72).

[5] Heat Transfer, M. Fishenden and O. A. Saunders. (Oxford University Press, London, 1950, p. 89).

[6] Method of Test for Normal Total Emittance of Surfaces of Materials 0.01 in. or less in Thickness at Approximately Room Temperature, ASTM Designation C445-59T.

[7] Measurement of Angular Emissivity, A. Umur, G. V. Parmelee and L. F. Schutrum (ASHAE Transactions, Vol. 61, 1955, p. 111).

[8] Tables of Thermal Properties of Gases (National Bureau of Standards, Circular 564, 1955).

[9] ASHRAE GUIDE, 1960, p. 108.

1765

No. 1765

Integrated Load Technique for Estimating Annual Energy Use of Central Air Conditioning Plants

A. EUGENE CONGRESS

Air conditioning service of the required integrity at the lowest total annual cost – that is our objective. However, when various choices are possible, it is sometimes difficult to decide which system design, granting equally acceptable integrity, will result in the lowest total annual cost. This paper describes how such a problem was approached and the results obtained.

The project involved selection of the refrigeration plant to air condition a 5-million-dollar addition to a large naval hospital. The initial capacity required was 900 ton with provision for future expansion to 2700 ton.

Preliminary studies revealed a sizable capital investment would be required to provide electrical power capacity for an all-electric refrigeration plant. Steam of sufficient capacity to supply the air conditioning was available year-round from the existing central heating plant. Building layout made it possible for steam to be brought into the refrigeration building merely by punching a hole in a wall of the adjacent heating plant. Since the ultimate plant will contain multiple units no stand-by capacity was considered necessary.

Five different schemes were evaluated:

Scheme A: 900-ton hermetic centrifugal compressor; electric motor drive; electric auxiliaries.

Scheme B: 125 psi steam to turbines driving the chilled water and condenser circulating water pumps; turbines exhausting at 12 psi to 900-ton absorption machine; electric motors drive cooling tower fans.

A. Eugene Congress is Head, Air Conditioning and Research Section, Power Generation Branch, Bureau of Yards and Docks, Department of the Navy. This paper was prepared for presentation at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961.

Scheme C: 900-ton centrifugal compressor, 125-psi steam turbine driver exhausting to 2.5 in. Hg condenser pressure; electric auxiliaries.

Scheme D: 900-ton lithium bromide absorption machine (12 psi steam); electric auxiliaries.

Scheme E: Split cycles: 125-psi steam turbine driving a 300-ton centrifugal compressor; turbine exhausting at 12 psi to a 600-ton absorption machine; electric auxiliaries.

Two aspects of the problem required special study: (1) determination of the initial capital investment and (2) estimating the annual operating cost. Estimates of procurement and installation costs were not difficult to obtain. But the comparative estimates of operating costs—especially energy costs—presented a more difficult problem. Preliminary estimates had been based on a simple rule-of-thumb solution. It assumed equivalent full-load operating hours for the refrigeration machine and total hours operation for the auxiliaries. It will be seen that this approach is not sufficiently accurate.

The major factor in evaluating relative annual operating costs is the energy charge, excluding for the moment the annual cost to amortize the capital investment. But how is energy to be charged?

DETERMINATION OF UNIT ENERGY COST

In deciding whether to purchase extra power or add equipment to generate power in a Navy-owned plant, the Navy determines the incremental or differential cost—the cost of each of the alternative schemes over and above the cost of existing level of operations. Costs of an existing plant, if they do not vary with the alternative considered, are not charged since they would have to be charged across the board and would have no effect on the outcome. Energy increments of steam and electricity will be considered in this study.

Steam—The principal incremental element of steam, as an energy source, is fuel. Whether a 50,000-lb/hr capacity boiler generates 25,000 lb/hr or 50,000 lb/hr, the difference in cost is determined by the amount of fuel consumed for the extra steam produced, plus the minor charges for additional utilities used in generation of the extra steam. These minor utility costs can be discounted when the stepped-up level of operations results in a compensating increase in boiler efficiency.

Electricity—The incremental unit cost of electric energy is the unit cost of that power block superimposed on existing consumption, and not the average annual kwhr cost for the whole customer power-consumption complex. For the plant under consideration, the cost consists of an energy charge in the 600,000 kwhr-per-month block, and a monthly kw demand charge for each month of operation.

ESTIMATING ANNUAL ENERGY AND WATER CONSUMPTION

Steps for determining the quantity

of energy, the kwhr or lb of steam consumed during an air-conditioning season, and makeup water for the cooling tower:

STEP 1: Determination of Climatic Data and Operation at Various Percentage Loads – The Weather Bureau of the United States Department of Commerce has compiled 5-year summaries of hourly observations of weather for each of 114 U. S. airway stations.* The number of hourly observations is indicated by 5-deg intervals for each month. Table I was compiled from such data for the Washington, D. C., area. It shows the average number of hours of dry-bulb temperatures for the air-conditioning months April-October in 5-deg intervals ranging from 60 to 104 F and the total number of hours in each of 8 temperature ranges.

Fig. 1 is a graphical presentation of the same data with the curve drawn through the mean of each temperature range. To the ordinate is added a percentage load scale (0 to 100) correlated directly to the 60 to 95 F plus temperature range. Dry-bulb temperature, rather than wet-bulb or other measurement, is assumed as more nearly representative of changes in cooling load. The energy consumption of alternative refrigeration schemes then was evaluated using this base of hours-temperature and percentage load.

STEP 2: Determination of Power Input – This involves the determination of power input to the refrigeration machine at the percentage loads corresponding to the mean temperature of each 5-deg temperature range. Power input of centrifugal compressors and absorption machines as a function of load is supplied by refrigeration equipment manufacturers; water rates (lb/bhp-hr) for steam turbine drives are obtained from turbine manufacturers. Fig. 2 shows the relationship between bhp and water rate to system load for the 970-bhp turbine drive used in the analysis of Scheme C.

STEP 3: Construction of Integrated

* ("Climatology of the U. S. No. 30," available from Superintendent of Documents, U. S. Government Printing Office, Washington, D.C.)

Table I—Hours of Temperature Occurrence per Month (Average of 5 Years: 1950-1955)

Temperature Range F	April	May	June	July	Aug.	Sept.	Oct.	Total Per Temp. Range
104-100			0.4	0.8				32
99- 95			7.2	15.0	4.6	4.4		
94- 90		0.8	27.4	44.8	16.0	10.0	1.6	101
89- 85	2.6	11.6	65.8	105.0	64.4	24.8	9.0	283
84- 80	13.6	34.2	96.5	159.0	120.0	54.8	16.2	494
79- 75	26.2	64.6	135.0	173.0	185.0	105.5	38.0	727
74- 70	42.2	109.0	165.0	166.5	223.0	159.5	98.5	964
69- 65	69.5	151.0	136.5	65.4	97.2	147.0	133.0	800
64- 60	104.0	155.0	58.8	12.2	28.3	121.0	159.0	638

Load Duration Curves – It is possible to estimate the total energy input for the chiller by multiplying the rate of energy input at various loads by the corresponding load hours. The energy input to auxiliary drives may be added, the amount depending upon the proposed method of operation of these units. However, the "Integrated Load Duration Curve," Fig. 3, is more illuminating and useful in complicated analysis.

The ordinate represents rate of energy input (kw, lb/hr steam, gal fuel oil, cu ft gas or Btu/hr). The abscissa represents the cumulative hours of operation at the various loads. The area under each step (such as ABCD) multiplied by the unit value of one sq in. gives the energy consumption for each partial load – for, kw × hr = kwhr or lb/hr steam × hr = lb steam, etc. Therefore, the total energy area under the steps multiplied by the unit area energy value yields the total energy input. The area under a smooth curve drawn through the midpoint of each riser is equivalent to the area under the steps. Energy consumed by auxiliary drives may be superimposed if of the same type of energy. A planimeter is used to measure the areas. Simi-

Fig. 1 Temperature and load variation vs. number hours (April-October)

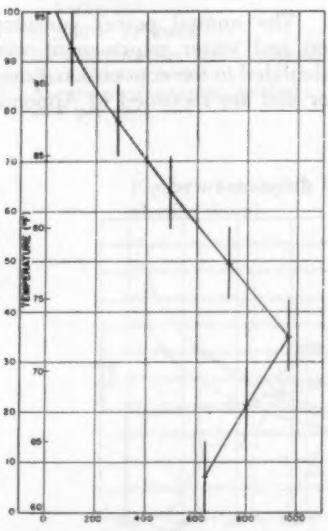
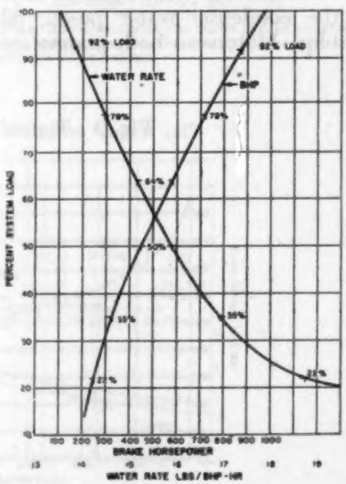


Fig. 2 970 hp – 7500 - rpm constant speed turbine. Bhp and water rate vs. system load



larly, a load duration curve may be plotted to obtain ton-hr refrigeration, designating the ordinate as tons.

INTEGRATED ENERGY AND INCREMENTAL COST PROCEDURES

Application of the integrated energy procedure to 2 of the 5 central-plant alternative designs will be presented herein, together with results obtained from the analysis of all 5 schemes. The data of Table II formed the base for the development of calculations involving power input.

Example I: Scheme A (All-Electric) Design Conditions: A 970-bhp motor drives a 900-ton centrifugal compressor. Leaving chilled-water temperature: 42 F. Condenser-water inlet temperature: 85 F. Fouling factor: 0.001. At 100 per cent system load the chilled-water pump takes 80 bhp; the condenser-water pump, 62 bhp. Motors on both pumps are

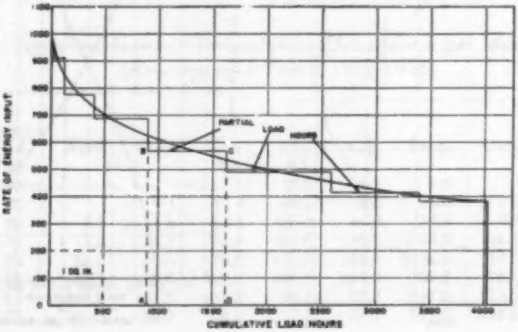
Table II—Per Cent Load at Mean Temperatures and Hours at Part Load Operation
(Developed from Table I and Fig. 1)

Temp. Range F	Per Cent Load Correlated to Mean of Temp. Range	Hrs
95+	100	32
94-90	92	101
89-85	78	283
84-80	64	494
79-75	50	727
74-70	35	964
69-65	22	800
64-60	7	638

two-speed – ½ speed on the chilled-water pump at 50 per cent, and below, system load; 2/3 speed on the condenser-water pump at 64 per cent, and below, system load. Two 20-hp motors drive cooling tower fans —but one on at 50 per cent load and below.

The annual power consumption and water requirement were calculated in the conventional manner and are recorded in Appendix

Fig. 3 Typical load duration curve



A. From these data the level of power input for compressor and auxiliaries, and ton refrigeration were used to plot the load duration curves of Fig. 4. The area under the lower full-line curve represents annual kwh consumption of the compressor motor; the area between the two full-line curves represents the annual power consumption of the motor-driven auxiliaries. Together these two areas indicate the total annual electrical consumption. The area under the dash-line curve represents the annual ton-hr refrigeration, which is a constant for each of the alternative schemes considered.

From this graph the following information is derived and compared for accuracy with calculated data in Appendix A:

(1) Total kwhr
$= 15.00$ in.^{2} total area $\times$ 100,000 kwhr/in.^{2}
$= 1.50 \times 10^6$ kwhr
By computation from Appendix A: 1,505,113 kwhr
Accuracy of graphic method: 99.7 per cent

(2) Compressor kwhr
$= 12.54$ in.^{2} $\times$ 100,000 kwhr/in.^{2}
$= 1.254 \times 10^6$ kwhr
By computation: 1,247,475 kwhr
Accuracy of graphic method: 99.5 per cent

(3) Equivalent Full-Load Energy Hours (Chiller)

$$= \frac{12.54 \text{ in.}^2 \text{ Compr. area}}{3.77 \text{ in. ordinate}} \times$$

500 hr/in.
$= 1663$ hr
By computation: 1655 hr
Accuracy: 99.52 per cent

(4) Refrigeration Ton-Hr
$= 14.26$ in.^{2} Refrig. area $\times$ 100,000 ton-hr/in.^{2}
$= 1.426 \times 10^6$ ton-hr
By computation: 1,425,042 ton-hr
Accuracy: 99.93 per cent

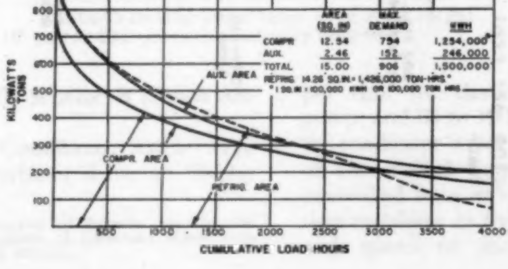
(5) Equivalent Full-Load Refrig. Hours

$$= \frac{1.426 \times 10^6 \text{ ton-hr}}{900 \text{ ton}}$$

$= 1584.4$ hr
By computation: 1583 hr
Accuracy: 99.91 per cent

(Note. The area under the curve can be equated to a rectangle using the length of the vertical ordinate as one side.)

Fig. 4 Scheme A load duration curve (all-electric drive)



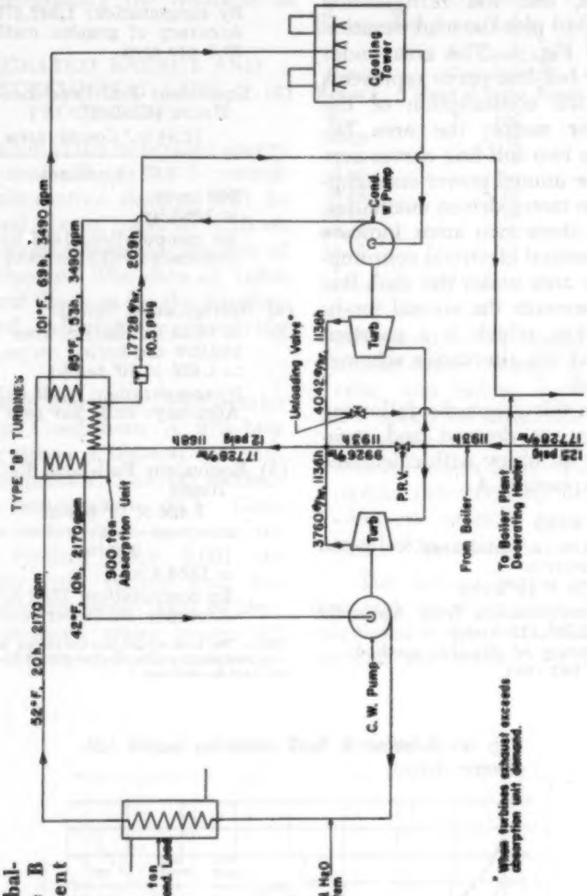


Fig. 5 Heat balance Scheme B (100 per cent load)

(6) Total Btu to Be Removed by Cooling Tower

$= $ System air conditioning load $+$ Compr. motor input

$= (1.426 \times 10^6$ ton-hr $\times 12{,}000$ Btu/ton-hr)

$+ (1.254 \times 10^6$ kwhr $\times 3{,}413$ Btu/kwhr)

$= 21.392 \times 10^9$ Btu

By computation: 21.36×10^9 Btu

Accuracy: 99.85 per cent

(7) Gal Water Loss at Cooling Tower

a. By Evaporation

$$= \frac{21.392 \times 10^9 \text{ (item 6)}}{1040 \text{ Btu/lb 95 F water vapor} \times 8.3 \text{ lb/gal}}$$

$= 2.478 \times 10^6$ gal

By comp: 2.4745×10^6 gal

Accuracy: 99.88 per cent

b. By Blowdown and Drift*

$$= \frac{2.478 \times 10^6 \text{ gal (item 7a)}}{5 \text{ concentration} - 1}$$

$= 0.6195 \times 10^6$ gal

By comp: 0.6186×10^6 gal

Accuracy: 99.85 per cent

Total Makeup $= 3.0975 \times 10^6$ gal

By computation: 3.093×10^6 gal

Accuracy: 99.86 per cent

(8) Gal Condenser Water Required to Remove Heat

$$= \frac{21.392 \times 10^9 \text{ Btu (item 6)}}{10 \text{ Btu/lb cooling tower temp. drop} \times 8.3 \text{ lb/gal}}$$

$= 257.73 \times 10^6$ gal

If condenser water were circulated full flow at all times during compressor operation, the annual quantity of water circulated would

$$= \frac{4039 \text{ hr } [(900 \text{ ton} \times 12000 \text{ Btu/ton}) + (754 \text{ kw} \times 3413 \text{ Btu/kwhr})]}{10 \text{ Btu/lb cooling tower temp. drop} \times 8.3 \text{ lb/gal}}$$

$= 650 \times 10^6$ gal, or 2.52 times that actually required.

Example II: Scheme B (Steam Absorption)

Design Conditions: Single-stage steam turbines drive an 80-bhp chilled-water pump and an 86-bhp condenser-water pump, both exhausting at 12 psig to 900-ton lithium bromide absorption chiller. Steam to turbine is supplied at 125 psig saturated from water-wall, air-preheater type boilers operating at 78 to 82 per cent efficiency. Estimated summer average efficiency 80 per cent. Boilers are fired with F.S. No. 6 fuel oil of 152,200 Btu/gal. There are three 30-hp motor driven fans on cooling tower. Two types of turbines are considered: one (designated on curves as Type A) is constant speed, direct connected; the other (designated on curves as Type B) is reduction geared, variable speed. Speed ranges 40 to 100 per cent for the chilled-water pump, and 64 to 100 per cent for the condenser-water pump. Speed of the chilled-water turbine is controlled by a differential pressure regulator in the coil system, and speed of the condenser-

* For determination of formula, see Fig. 151 in "Betz Handbook of Industrial Water Conditioning," 5th Edition.

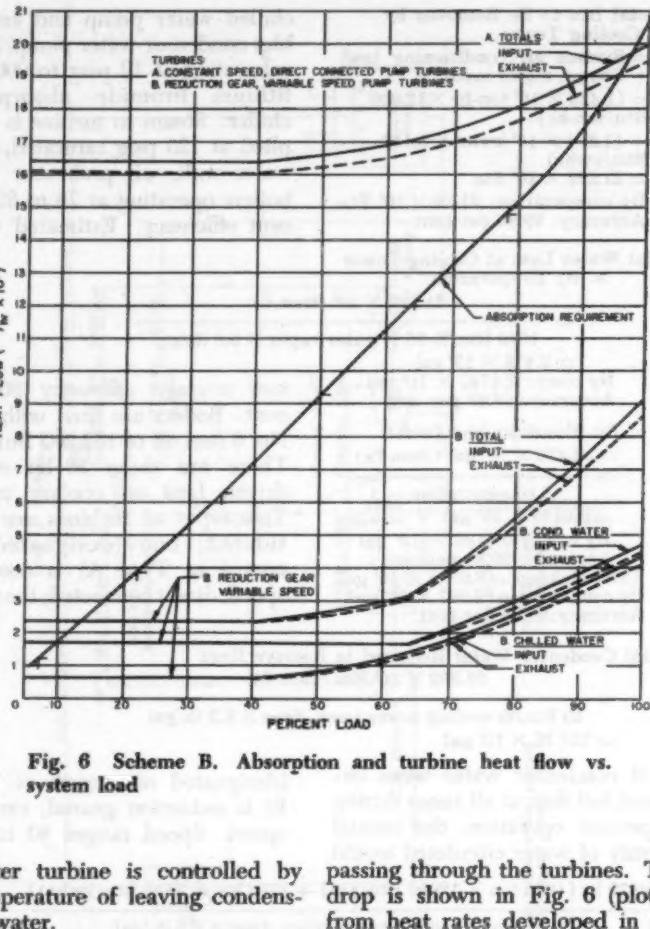


Fig. 6 Scheme B. Absorption and turbine heat flow vs. system load

water turbine is controlled by temperature of leaving condenser water.

The number of hours of operation is the same at the 8 partial system loads as in the previous all-electric example.

Fig. 5 shows the diagrammatic and heat balance at full load when utilizing high-speed, reduction-gear type turbines. The steam suffers a small thermal drop while passing through the turbines. This drop is shown in Fig. 6 (plotted from heat rates developed in Appendices B and C) where the difference in heat flow between turbine inlet and exhaust is the thermal drop across the turbines. While the turbine drop in passing from maximum load to minimum load is fairly constant for Type A turbines, for Type B turbines the drop is so small at loads below 40 per cent

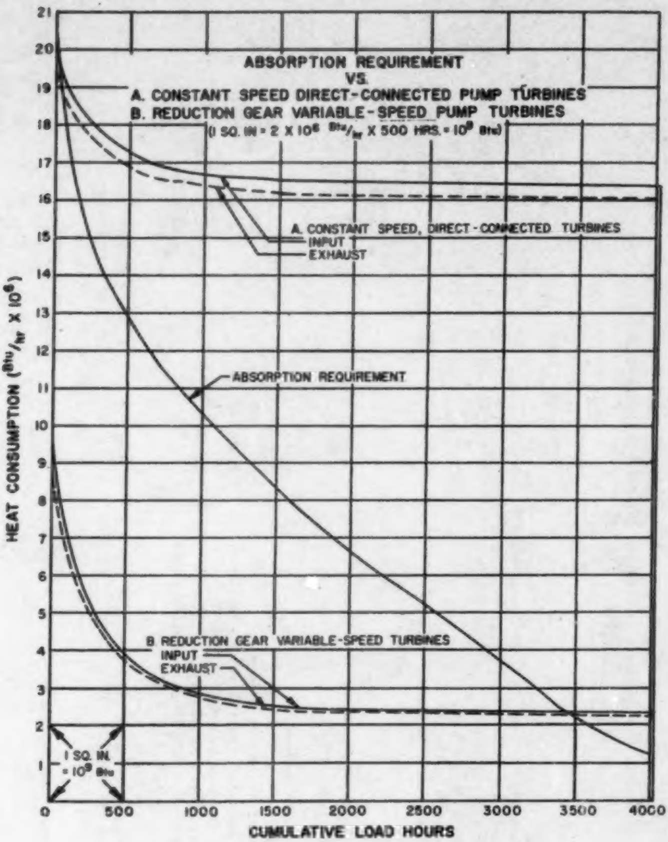


Fig. 7 Load duration curves for Scheme B

that it cannot be indicated clearly on the scale of this graph. This reduction in turbine drop stems from the fact that the thermal loss across a turbine is inversely proportional to its water rate—lb/bhp hr, the latter increasing at lower loads (see Appendix B).

Fig. 6 also shows the relationship between the rate of heat input required by the absorption unit (developed from Appendix D) and the heat exhausted from the turbines. For the most part the heat flow per hour from Type A turbines is too great for the needs of absorption; for Type B, too small.

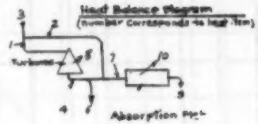
The load duration curves of

Table III—Scheme B: Annual Steam Heat Balance (From Fig. 7)

Design Conditions

Pressure steam to turbine	125 psig
Pressure steam to absorption	12 psig
Enthalpy 125 psig steam (saturated)	1193 Btu/lb
Enthalpy 12 psig steam (saturated)	1162 Btu/lb
Enthalpy saturated liquid from absorption unit	209 Btu/lb

A—Constant-speed, direct-connected turbines
B—Reduction gear, variable-speed turbines



Heat Item	Heat	CURVE (See Fig. 7) A — Area in sq. in. or 10^9 Btu (except as noted)	B
1	Input to turbines	67.70	11.86
2	PRV makeup to absorption	0.02	18.30
3	Total from boiler (Item 1 + Item 2)	67.72	30.16
4	Exhaust from turbine	66.25	11.38
5	Drop across turbine (Item 1 — Item 4)	1.45	0.48
6	Excess turbine exhaust	36.89	0.30
7	Absorption input (Item 2 + Item 4) — Item 6	29.38	29.38
8	Equivalent lb 12 psig saturated steam to absorption (Item 7 ÷ 1162 Btu/lb)	25.28 ×	10^6 lb
9	In absorption leaving condensate (Item 8 × 209 Btu/lb)	5.28	5.28
10	Drop across absorption (Item 7 — Item 9)	24.10	24.10
11	Chargeable to refrigeration system (Item 5 + Item 10)	25.55*	24.58*
12	Balance (Item 3 = sum of Items 5, 6, 9, 10)		

Excess of curve "A" over curve "B":

Boiler steam (67.72-30.16) 10^9 = 37.56 × 10^9 Btu = 31.5 million lb 125 psig saturated steam
Excess turbine exhaust (36.89-0.30) 10^9 = 36.59 × 10^9 Btu = 31.5 million lb 12 psig saturated steam
Turbine drop (1.45-0.48) 10^9 = 0.97 × 10^9 = 813 thousand lb 125 psig saturated steam

---
* If full use can be made of excess turbine exhaust

Fig. 7 (plotted from Fig. 6) indicate what the situation will be on an annual basis. The considerable excess of Type A exhaust over absorption requirement will be unwelcome in a plant where it cannot be utilized profitably. The condition of Type B exhaust is less objectionable, since the area of deficiency between exhaust and absorption requirement is made up by PRV steam. The area of excess steam is so small as to be utilized for boiler feedwater heating.

Note that the area on the graph between turbine input and exhaust, representing the turbine thermal drop, is the amount of energy necessary to drive the chilled-water and condenser-water pumps the year round. In the case of the Type B geared, variable speed turbine, it is so small that if disregarded it would be well within estimating accuracy. Only where the final evaluations of alternative choices show a "photo finish" should one be concerned with the thermal drop across auxiliary turbine drives.

The heat balance of Fig. 7, recorded in Table III, shows the marked superiority of a high speed, reduction gear, variable rpm turbine over the constant speed, direct-connected turbine. The latter takes approximately 31.5 million lb boiler steam more than the reduction-gear, variable speed job, and poses the problem of what to do with an almost equal quantity of surplus exhaust. When, as in this case, the by-product steam cannot be utilized profitably, use of the more expensive geared turbine is justified since the initial extra cost can be amortized in a relatively short time. Also, the variable speed feature permits rendering service in accordance with demand, and avoids wasteful pumping.

The following significant information is derived from the data of Table III on an annual basis for refrigeration Scheme B using the reduction-gear Type B turbine:

(1) Fuel Oil Chargeable to Refrigeration System

$$= \frac{24.58 \times 10^9 \text{ Btu (item 11 of Table III)}}{152{,}200 \text{ Btu/gal} \times 0.80 \text{ boiler efficiency}}$$
$$= 201{,}873 \text{ gal} = 4806 \text{ bbl}$$

(2) Equivalent Full-Load Energy Hours (Chiller)

$$= \frac{29.38 \times 10^9 \text{ (item 7 of Table III)}}{20.8 \times 10^6 \text{ Btu (100 per cent absorption heat from Fig. 6)}}$$
$$= 1418 \text{ hr}$$

(3) Total Heat to Be Removed by Cooling Tower

$$= (\text{ton-hr} \times \text{Btu/ton-hr}) + \text{chiller steam-heat drop}$$
$$= (1.426 \times 10^6 \text{ ton-hr}^*) (12000 \text{ Btu/ton-hr})$$
$$+ 24.10 \times 10^9 \text{ Btu (item 10 of Table III)}$$
$$= (17.112 + 24.10)\, 10^9 = 41.212 \times 10^9 \text{ Btu}$$

* (from Example I, item 4)

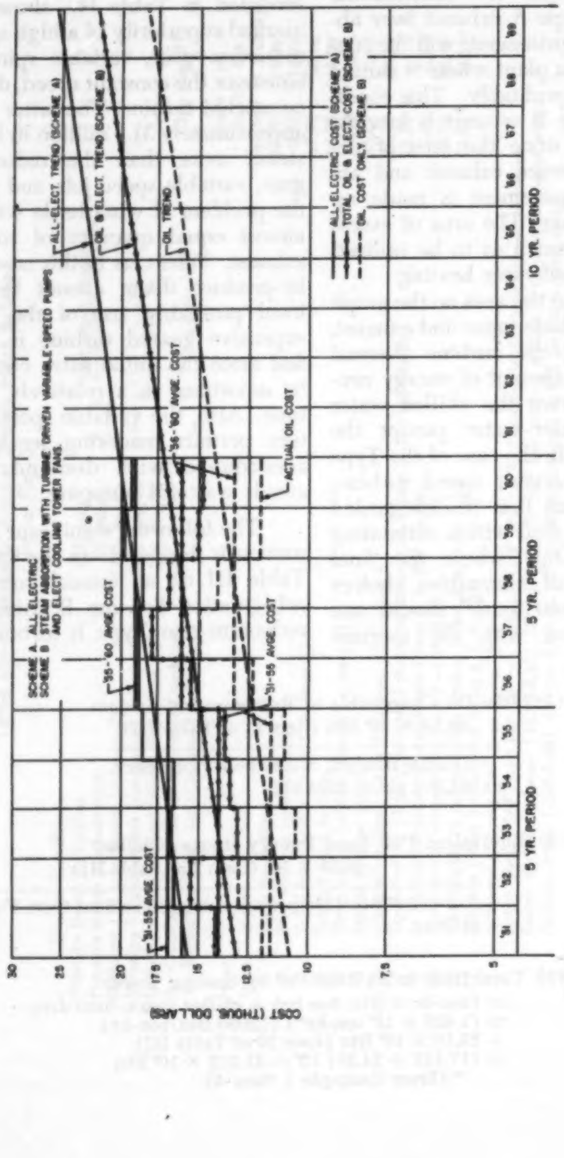


Fig. 8 Comparative evaluation of projected energy costs

(4) Gal Water Loss at Cooling Tower

a. By Evaporation

$$= \frac{41.212 \times 10^9 \text{ Btu total heat}}{(1036 \text{ Btu/lb for 101 F water}) (8.3 \text{ lb/gal})}$$

$$= 4.79 \times 10^6 \text{ gal}$$

b. By Blowdown and Drift

$$= \frac{4.79 \times 10^6 \text{ gal}}{5 \text{ concentration} - 1}$$

$$= 1.20 \times 10^6 \text{ gal}$$

$$\text{Total Makeup} = (4.79 + 1.20)\, 10^6$$

$$= 5.99 \times 10^6 \text{ gal}$$

(5) Condenser Water Required to Remove Heat

$$= \frac{41.212 \times 10^9 \text{ Btu removed by tower}}{(16 \text{ Btu/lb drop} \times 8.3 \text{ lb/gal})}$$

$$= 310 \times 10^6 \text{ gal}$$

If condenser water were circulated full-flow at all times during absorption operation, the quantity circulated would

$$= 4039 \text{ hr} \times 3490 \text{ gpm (from Fig. 5)} \times 60$$

$$= 846 \times 10^6 \text{ gal}$$

or 2.7 times that actually required.

DETERMINATION OF INCREMENTAL ENERGY COST (ANNUAL BASIS)

As mentioned earlier, in determining the cost of operating a new facility, the Navy employs the incremental rate principle – the cost of that block of new power over and above the existing usage. In the competition between producers of electricity and other types of energy supply, the Navy is neutral. Its prime interest is to discharge its responsibility to taxpayers by using in each application the most economically available energy–all factors considered.

If current unit prices of electricity and fuel oil are used, criticism might be leveled by one supplier or another if its price happened to be momentarily high unless supported by price trend information. It was therefore thought advisable to see what effect the incremental energy load would have if superimposed on the then existing loads of the past decade. What would have been the incremental annual cost for the years 1951 through 1960? Fig. 8 shows this for the all-electric Scheme A and the absorption Scheme B, based on actual unit incremental cost to the government.

Although the price of oil during that decade fluctuated more than that of electricity (due to the Korean War and the Suez crisis), the graph shows that the trend in energy costs of the two schemes moved side by side. If it can be assumed that this trend will continue in the future, the relative dif-

Table IV—Comparative Annual Energy and Water Operating Cost

		Scheme A	Scheme B	Scheme C	Scheme D	Scheme E
Design Conditions:			Steam Pump Drivers Exhaust	Steam Turbine		Steam Turb.
42 F leaving chilled water 85 F inlet condenser water 0.001 fouling factor 4039 total operating hr		All Electric Refrigerant System	to Absorption Chiller Elec. Fans Type B Turbine	Driven Refrigerant Compr. Elec. Aux.	Straight Absorption Chiller Elec. Aux.	Driven Compr. Exhausts to Absorption Chiller
Quantities						
Electricity	Units					
Energy	kwhr	1,505,113	249,231	430,611	452,877	332,021
Demand						
5 month at	kw	906	99	242	236	178
2 month at	kw	714	99	238	232	174
Fuel oil (F.S. No. 6)	bbl		4,806	5,624	4,806	4,840
Water makeup	10^3 gal	3,093	5,960	6,640	5,960	6,085
Utility Unit Rates (1960)						
Electricity		$	$	$	$	$
Energy	kwhr	0.00731	0.00731	0.00731	0.00731	0.00731
Demand	kw	1.40	1.40	1.40	1.40	1.40
Fuel oil (F.S. No. 6)	bbl		2.516	2.516	2.516	2.516
Water	10^3 gal	0.41	0.41	0.41	0.41	0.41
Water treatment	10^3 gal	0.03	0.03	0.03	0.03	0.03
Costs (1960)						
Electricity						
Energy		$11,002	$ 1,822	$ 3,148	$ 3,311	$ 2,427
Demand (5 month)		6,342	693	1,694	1,652	1,246
Demand (2 month)		2,000	277	666	650	487
Total Electricity		19,344	2,792	5,508	5,613	4,160
Fuel			12,092	14,150	12,092	12,177
Electricity and fuel		19,344	14,884	19,658	17,705	16,337
Water makeup		1,268	2,444	2,722	2,444	2,495
Water treatment		93	179	199	179	183
Total water		1,361	2,623	2,921	2,623	2,678
Total annual energy and water cost		$20,705	$17,507	$22,579	$20,328	$19,015

ference in the energy operating costs of the 5 schemes, listed in Table IV and shown graphically in Fig. 9, will also continue.

CAPITAL VALUE OF ANNUAL SAVINGS

While the comparative annual operating costs of Table IV show an annual saving of $5100 for the lowest cost scheme when compared with the highest, this does not by itself justify the installation of the lowest operating cost design. The installed cost of one scheme may be so much less than another that the difference in annual amortized cost would more than offset the savings. An annual savings is equivalent to an annual annuity and can be converted to its corresponding equivalent capital investment, called present worth. The necessary factors for performing this calculation are provided in tables of investment, compound interest and annuity.

The present worth factor at 4 per cent annual compound interest (the rate the government pays for long-term money today) over a 15-year equipment life is 11.118. (This factor will vary, depending upon the life of the amortized equipment and the interest rate the owner must pay for his money.) The present worth calculation then is:

$$\$5,100 \times 11.118 = \$56,700$$

This means that to be competitive the total installed first cost of the turbine-driven centrifugal machine would have to be $56,700 less than that of equipment of Scheme B. And the motor-driven centrifugal, to be competitive, would have to be $35,600 less than Scheme B.

USE OF EQUIVALENT FULL-LOAD HOURS

This study has revealed that results are likely to be misleading when the same number of equivalent full-load hours is applied to estimate annual energy consumption of alternative types of chillers for the same refrigeration job.

Two different concepts are involved: (1) the output of the plant measured in equivalent full-load refrigeration hr; and (2) the input measured in equivalent full-load energy hr.

(1) Equivalent full-load refrigeration hr

$$= \frac{\text{summation of ton-hr at the various loads}}{\text{refrigeration tons at 100 per cent load}}$$

and is a constant for the same refrigeration job regardless of the type of chiller equipment and kind of energy used. It is the number of hours the chiller operates continuously at 100 per cent load to produce the annual refrigeration effect.

Equivalent full-load energy hours, on the other hand, is not necessarily a constant for alternative chiller equipment considered for an installation.

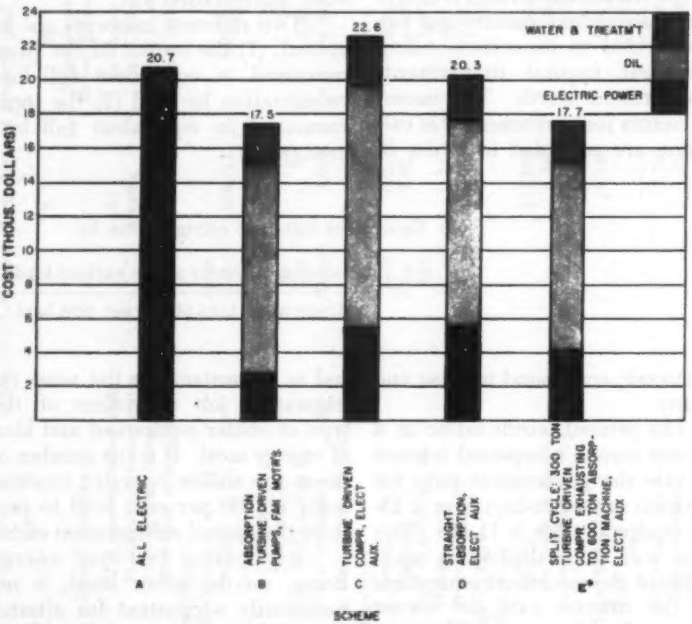
(2) Equivalent full-load energy hr

$$= \frac{\text{summation of energy consumed at various loads}}{\text{rate of energy input at 100 per cent load}}$$

and is the number of hours the chiller would operate continuously at 100 per cent system power supply to equal its annual energy consumption.

In the present study equivalent full-load refrigeration hr came to 1583 hr – constant for all of the 5 schemes considered. Equivalent full-load energy hr, developed from duration curves (or computation when feasible), is different for different chiller equipment as shown by Table V. If, for example, the 2089 equivalent full-load energy hr of Scheme C were used to estimate annual energy consumption of the other schemes, would not an injustice be done? To take another example: if the 1418 hr of the absorption chiller (Schemes C and D) were applied to the peak compres-

Fig. 9 Comparative annual electric, oil and water cost



* To correspond with later revisions to Scheme E of Table IV, the oil portion of the Scheme E column should be raised to a level of approximately 16.3 and the top of the water portion to 19.0.

sor kw demand to estimate annual energy consumption of the motor-driven centrifugal, the result would be low by 17 per cent.

Also, the equivalent full-load energy hours derived by analysis for one job may not be interchangeable with that on another – even using the same type of equipment. Examination of Equation (2) shows that should the chiller operate at a higher or lower system load for a different number of hours—causing a change in the quantity of energy consumption—and thus increase or decrease the numerator, the resulting equivalent full-load energy hours could change. Therefore, one cannot use equivalent full-load energy hours as a short-cut to estimate annual energy consumption unless two conditions are satisfied: (a) the chiller is of the same type and (b) the conditions of operation are identical.

INCREMENTAL BOILER EFFICIENCY

What is a reasonable and justifiable boiler efficiency to use when estimating fuel required to furnish steam for a steam-actuated refrigeration machine? The boilers in the hospital for which this study was made are of the air-preheater water-wall type. Efficiency of operation at half load is 78 per cent, and at full load, 82 per cent. Adding a new summer air conditioning load permits operation at higher than existing summer loads. What will be the efficiency of the added load?

Consider a boiler generating 50,000,000 Btu/hr. The input at 78 per cent efficiency is 50,000,000 Btu/hr ÷ 0.78 = 64,100,000 Btu/hr. If the load on this boiler is increased to 100,000,000 Btu/hr at 82 per cent efficiency, the input is 100,000,000 Btu/hr ÷ 0.82 = 122,000,000 Btu/hr. Thus for an output increment of 50,000,000 Btu/hr the incremental input is (122,000,000–64,000,000) 57,900,000 Btu/hr. This is equivalent to 50,000,000 Btu/hr ÷ 57,900,000 Btu/hr = 86.4 per cent efficiency. We thus realize a better efficiency for the added increment than for the overall operation.

The foregoing incremental relationship may be expressed as follows:

$$\frac{50}{0.78} + \frac{50}{X} = \frac{100}{0.82}$$

where X = 86.4 per cent

In estimating fuel consumption for steam-actuated equipment in the 900-ton study, an efficiency of 80 per cent is used instead of the 86.4 per cent indicated above. This makes allowance for the small extra power consumed by fans and pumps required in boiler feed-water, fuel and fan-system auxiliaries of boiler operation.

Table V—Equivalent Full-Load Chiller Energy Hours

Scheme		Hr	Percent in Excess of Scheme B
A	All-Electric	1655	17
B	Steam Turbine-Driven Pumps Exhaust to Absorption	1418	0
C	Steam Turbine Driven Compressor	2089	47
D	Straight Absorption	1418	0
E	Split Cycle	1887	33

ANNUAL FIXED CHARGE OF BOILER PLANT

Should the annual fixed cost of that portion of an existing boiler plant used for steam air conditioning be charged to the scheme using steam power? The Navy holds that when an existing boiler plant (as in the case under consideration) has ample boiler capacity and requires no expansion – entailing no significant extra operating cost except fuel to supply steam – a charge for interest, depreciation and insurance on the boiler plant is not justified. The steam plant is there, and will be there, regardless of the type of refrigeration equipment installed.

CONCLUSION

The techniques, procedures and approach described herein present a method of evaluating accurately air conditioning refrigeration plant designs for purpose of equipment selection. In the specific application to a Naval 900-ton refrigeration project, the capital worth of annual savings justified the selection of a steam system. The question of maintenance was carefully considered. After weighing all the factors, it

Appendix B: Scheme B: Pump Turbine Thermal Calculations for Type "B"*

Per cent System Load	BHP	Water Rate lb/bhp hr	Steam Rate lb/hr (Col 2 × Col 3)	Heat Input Btu/hr (1193 Btu/lb × Col 4)	Turbine Heat Drop Btu/lb $\left[\frac{2545 \text{ Btu/bhp hr}}{0.95 \times \text{Col } 3}\right]$	Turbine Heat Drop Btu/hr (Col 4 × Col 6)	Heat in Exhaust Btu/hr (Col 5 — Col 7)
1	2	3	4	5	6	7	8
					80-hp chilled water pump turbine drive		
100	80	47	3,760	4,485,680	57	214,320	4,271,360
92	62	47.5	2,945	3,513,385	56	164,920	3,348,465
78	39	52.5	2,048	2,443,264	51	104,448	2,338,816
64	20	61	1,220	1,455,460	44	53,680	1,401,780
50	10	76.5	765	912,645	35	26,775	885,870
≤40	6	90	540	644,220	30	16,200	628,020
					86-hp condenser water turbine drive		
100	86	47	4,042	4,822,106	57	230,394	4,591,712
92	70	49	3,430	4,091,990	55	188,650	3,903,340
78	43	55	2,365	2,821,445	49	115,885	2,705,560
≤64	23	64	1,472	1,756,096	42	61,824	1,694,272

* Reduction gear, variable speed turbines

was held in this specific project that the difference in maintenance required for the various schemes was negligible.

However, it is conceded that the selection, which was the most economical and logical under the conditions surrounding this installation, may not be the same for other installations having different governing conditions. Much depends upon the factors of availability and type of power; the incremental cost of fuel to make steam, electricity and water, the steam rate of selected turbines; profitable use of by-product steam and heat of condensate; applicable taxes, interest and insurance on capital investment.

The Navy proceeds upon the assumption that every effort should be made to insure the maximum return for the taxpayers' dollars. An economic analysis of air-conditioning operating costs as herein described is required. The procedures demonstrated should prove helpful in guiding contracting engineers to provide evaluations in accordance with these criteria: Service of the required integrity at the lowest total annual cost.

	Annual Heat (Btu)		
	Input	Drop	Exhaust
Hr	(Col 5 × Col 9)	(Col 7 × Col 9)	(Col 10 — Col 11)
9	10	11	12
32	143,541,760	6,858,240	136,683,520
101	354,851,885	16,656,920	338,194,965
283	689,000,448	29,454,336	659,546,112
494	718,997,240	26,517,920	692,479,320
727	663,492,915	19,465,425	644,027,490
2,402	1,547,416,440	38,912,400	1,508,504,040
	4,117,300,688	137,865,241	3,979,435,447
32	154,307,392	7,372,608	146,934,784
101	413,290,990	19,053,650	394,237,340
283	798,468,935	32,795,455	765,673,480
3,623	6,362,335,808	223,988,352	6,138,347,456
	7,728,403,125	283,210,065	7,445,193,060

Appendix A: Electric Power and Water Calculations for Scheme A

	Column→	1	2	3	4	5	6	7	8	9
		Per Cent System Load								
Compressor (970 bhp)		100 per cent	92 per cent	78 per cent	64 per cent	50 per cent	35 per cent	22 per cent	7 per cent	Total
1. Refrigeration	Ton	900	828	702	576	450	315	198	63	
2. Compr. input as per cent full load	Per Cent	100	90	75	60	48	35	26	20	
3. Compr. input (line 2 × line 3 col 1)	kw	754	679	566	453	362	264	196	151	
4. Hours of operation	Hr	32	101	283	494	727	964	800	638	4,039
5. Electrical consumption (line 3 × line 4)	Kwhr	24,128	68,579	160,178	223,782	263,174	254,496	156,800	96,338	1,247,475
6. Ton-hr (line 1 × line 4)	Ton-Hr	28,800	83,628	198,666	284,544	327,150	303,660	158,400	40,194	1,425,042
7. Equiv. full load energy hour (line 5 col 9 ÷ line 3 col 1)	Hr									1,655
8. Equiv. full load refrig. hour (line 6 col 9 ÷ line 1 col 1)	Hr									1,583
Auxiliaries										
9. Chilled water pump (80 BHP)[1]	kw	67	66	63	60	10	10	10	10	
10. Condenser water pump (62 BHP)[2]	kw	52	52	52	18	18	18	18	18	
11. Cooling tower fans (2-20 HP)[3]	kw	33	33	33	33	17	17	17	17	
12. Total aux. elec. demand	kw	152	151	148	111	45	45	45	45	
13. Total aux. elec. consumption (line 4 × line 12)	kwhr	4,864	15,251	41,884	54,834	32,715	43,380	36,000	28,710	257,638
Compressor and Auxiliaries										
14. Total compr. and aux. demand (line 3 + line 12)	kw	906	830	714	564	407	309	241	196	
15. Total elec. consumption (line 5 col 9 + line 13 col 9)	kwhr									1,505,113
Cooling Tower Duty (Annual Basis)										
Btu to be removed										
16. a. Air conditioning load (line 6 col 9 × 12000 Btu/ton hr)							10^9 Btu	17.10		
17. b. Power input (line 5 col 9 × 3413 Btu/kwhr)							10^9 Btu	4.26		
18. Total (Sum line 16 and 17)							10^9 Btu	21.36		
Gal water makeup										
19. a. Loss by evaporation (line 18 ÷ [(1040 Btu/lb for 95 F vapor) × 8.3 lb/gal)]							10^6 Gal	2.4745		
20. b. Loss by blow down & drift (line 19 ÷ (5 cycles chemical conc. — 1)							10^6 Gal	0.6186		
21. Total (sum lines 19 and 20)							10^6 Gal	3.0931		

[1] Two speed motor, ½ speed ≤ 50 per cent load
[2] Two speed motor, ⅔ speed ≤ 64 per cent load
[3] One fan on ≤ 50 per cent load

Appendix C: Pump Turbine Thermal Calculations for Type "A"*

Per cent System Load 1	BHP 2	Water Rate lb/bhp hr 3	Steam Rate lb/hr (Col 2 × Col 3) 4	Heat Input Btu/hr (1193 Btu/lb × Col 4) 5	Turbine Heat Drop Btu/lb $\frac{2545 \text{ Btu/bhp hr}}{0.95 \times \text{Col.3}}$ 6	Turbine Heat Drop Btu/hr (Col 4 × Col 6) 7	Heat in Exhaust Btu/hr (Col 5 — Col 7) 8
80 hp chilled water pump turbine							
100	80	94	7,520	8,971,360	28.5	214,320	8,757,040
92	79	94	7,426	8,859,218	28.5	211,641	8,647,577
78	75	94	7,050	8,410,650	28.5	200,925	8,209,725
64	70	94	6,580	7,849,940	28.5	187,530	7,662,410
50	66	94	6,204	7,401,372	28.5	176,814	7,224,558
40	63.5	94	5,969	7,121,017	28.5	170,116	6,950,901
86 hp condenser water pump turbine							
100	86	108	9,288	11,080,584	25	232,200	10,848,384
92	81	108	8,748	10,436,364	25	218,700	10,217,664
78	74	108	7,992	9,534,456	25	199,800	9,334,656
64	72	108	7,776	9,276,768	25	194,400	9,082,368

* Constant speed, direct-connected turbines

Appendix D: Absorption Heat Input*
Steam at 12 PSIG Saturated (1162 Btu/lb)

Per cent System Load 1	Refrig. Ton 2	Hours Operation 3	lb/ton hr 4	Steam Rate lb/hr (Col. 2 × Col. 4) 5	Heat Rate Btu/hr (Col. 5 × 1162 Btu/lb) 6	Total Heat Input 10^6 Btu (Col. 3 × Col. 6) 7
100	900	32	19.8	17,820	20,706,840	662.62
92	828	101	18.8	15,566	18,087,692	1,826.86
78	702	283	18.5	12,987	15,090,894	4,270.72
64	576	494	18.2	10,483	12,181,246	6,017.54
50	450	727	17.8	8,010	9,307,620	6,766.64
35	315	964	17.2	5,418	6,295,716	6,069.07
22	198	800	16.8	3,326	3,864,812	3,091.85
7	63	638	16.6	1,046	1,215,452	775.46
		4,039			TOTAL/YR	29,480.76

* Two 450-ton machines in parallel

DISCUSSION

LEIGH ST. JOHN, Binghamton, N. Y.: Does this paper contain a tabulation of the installation costs in the various schemes?

AUTHOR CONGRESS: In this paper I estimated the annual energy cost. The tabulated investment cost is available, however.

L. ST. JOHN: In order to arrive at your figures, is it necessary to have the capital cost also?

AUTHOR CONGRESS: The capital cost certainly must be included. In this study the capital cost of putting in the turbine driven centrifugal compressor would have to be $56,700 less than the capital cost of Scheme B to be competitive.

L. ST. JOHN: This specific type of installation fitted a specific job which you had and you were only going to use your steam boilers for processed load, I assume, during this summer time.

AUTHOR CONGRESS: Yes, we have a large boiler plant there, and only a small fraction of it is used during the summer. There would be no additional cost of operating the plant for the same personnel are there; the difference would be in the fuel. This is shown in the appendices. Detail computations are included only for Schemes A and B; because of the limitation of the paper, more information could not be added.

We have found, in preparing the accuracy of the values obtained under the graphs with the computations, that the graphs are 99.5% or plus accurate.

All that one would have to do is determine the percentages of system load, operating hours corresponding to the percentages of system load; find energy input per hour and make two graphs. One is power or rate of energy input vs. per cent system loads; the second, load duration which shows power input vs. accumulated load hours. The areas under the latter curves represent annual energy consumption.

ROBERT FLANAGAN, West Des Moines, Iowa: The problem that we encounter is that we can estimate the equivalent full load hours, refrigerating hours, that we might anticipate for a certain type of structure. It would appear that in order to utilize the information you have presented, it must be converted back into a graph form such as you have presented. Is there any way of converting equivalent full load refrigeration hours into these two graph forms? If so, would this change the equivalent full load hour estimate? By changing that, the entire machine selection or type of system that will be installed might be changed.

AUTHOR CONGRESS: The answer to the second question is yes, you may have to change the entire evaluation of the system to be installed. You cannot determine your equivalent full load hours by taking it out of a book. Each specific case, each job has different equivalent full load energy hours. This procedure tells you what they are. As shown by Table V, they differ for each different design.

L. ST. JOHN: A specific case has been picked here on this possible load which is advantageous towards steam-driven equipment of some kind. Has an analysis been made over a break-even point? You have an actual steam load under summer operation, which must be maintained. Have you ever made an analysis on other types of operations other than those which require heavy steam loads in the summer time?

AUTHOR CONGRESS: In all our Navy installations, we have steam. The Naval Shore Establishment has about 18,000 boilers; about 14,000 operating at one time and we always have steam there. Our problem is to determine whether we should put in a steam-driven job, use the steam we have or in some cases high temperature water, or an electric job. Now, since this technique was developed, we are thinking of applying it to all large central air conditioning plants.

L. ST. JOHN: If it were not economical to operate your boiler plant in the summertime, it would be cheaper to shut it down completely. I am assuming that some of your places must have separate heating loads and air conditioning loads; therefore, you have a condition that depends on many things—whether it is coal-fired, whether it is automatically gas-fired or oil-fired or something else.

AUTHOR CONGRESS: If we do not operate a boiler plant in the summer, we have to evaluate the scheme on the basis of electricity. There does not seem to be any alternative without steam or high temperature water at this time. We are also trying to evaluate the split cycle, where you use the steam turbine to drive a centrifugal compressor, turbine exhausting to an absorption machine. We have been advised by industry to use a one-third, two-third split. In this case, 300 ton on a turbine driven compressor and 600 ton on an absorption. This is satisfactory if most of your hours are at 100% load; it comes out to about 14.4 lb of 125 psig steam per ton at 100% load, the most efficient point, corresponding to only 32 hours. When the load goes down, the rate of steam consumption is higher. We also evaluated the effect of another split—200 ton centrifugal and 700 ton absorption. It was found that at this split there are more hours of series operation. The idea is to use the series or dual cycle during the maximum hours of operation. It was found that by using 200 ton on the centrifugal and 700 on the absorption about 4 million lb of steam per annum can be saved. This gets down to a rate of about 13½ lb steam per ton at the most efficient point over a longer number of

operating hours. My advice is to make a study and determine where the maximum hours of operation are and then select the tonnage split between centrifugal and absorption so both machines will be in operation during those hours.

If you save 4 million lb of steam in a different split, and assuming that your steam costs $1 per 1000 lb, there is a $4000 savings per year. Present capital worth of a $4000 annual savings at 4% interest comes to about $44,000. In other words, on this savings item alone, one could spend $44,000 more in first cost and still be competitive. In the split system we plan to evaluate the use of the gas turbine to drive the centrifugal and the exhaust gas for use in a boiler to generate steam or high temperature water to power the absorption. An alternative scheme is to use diesel drive and the exhaust to make steam for the absorption. There are various possible combinations.

1766

No. 1766

Joint ASHRAE-ASME Meeting
November 28-29, 1961
New York, New York

A new psychrometric chart, the most significant such development since the publication of Dr. Willis H. Carrier's Rational Psychrometric Formulae in November, 1911, was revealed by ASHRAE at a joint session between this Society and ASME on November 28 in New York. This session, co-sponsored by ASHRAE and the Process Industries Div of ASME, took place during the latter society's annual meeting.

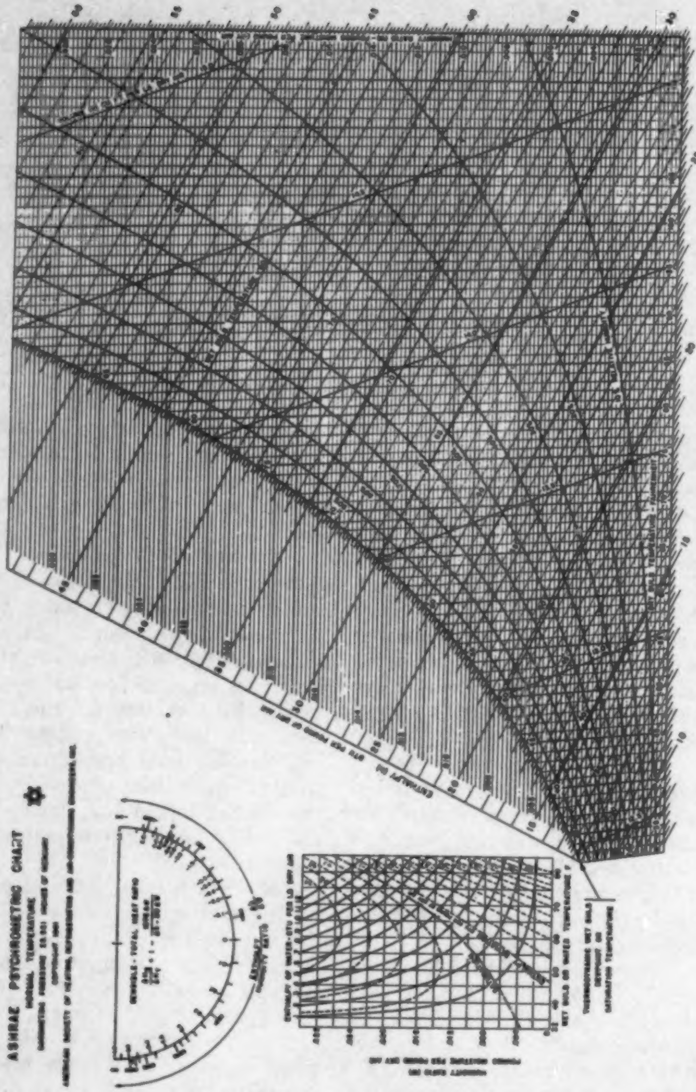
The new chart shows the degree of moist or dry air as determined from the properties of temperature, heat content, moisture content and air volume. The chart is invaluable to engineers in solving the problems requiring the conditioning of air.

Presented by ASHRAE President John Everetts, Jr., the new chart is designed to replace the previous charts adopted by ASHRAE's predecessor societies. The new psychrometric development is basically an enthalpy-humidity ratio chart on which factors such as dry bulb, wet bulb, dew point and volume information are evident.

Evolved by ASHRAE's Psychrometric Panel, the chart, reproduced upon an adjacent page, uses as a coordinate system enthalpy and humidity ratio, thus a mixture is on the straight line joining the two individual state points. The wet bulb lines are straight lines and the dry bulb lines are nearly straight lines but spreading out toward the top of the chart. The 120 F dry-bulb line is normal to the base line.

Since a normal starting point for the solving of problems uses the dry bulb and wet bulb, these functions together with relative humidity and humidity ratio are shown as a complete grid of lines.

In the interest of keeping the chart "clean," a skeleton grid of enthalpy lines has been shown in



This is the new official ASHRAE Psychrometric Chart

ASHRAE President John Everetts, Jr., presents a scroll commemorating the 50th anniversary of Dr. Willis H. Carrier's publication of "Rational Psychrometric Formulae" to Cloud Wampler, center. At right is William H. Byrne, President of ASME

the working area of the chart, but to provide a working tool, complete scales of enthalpy are shown at both ends of the working space, thus making it possible to locate the enthalpy accurately for any state point by the use of a ruler or straight-edge.

Approximate values of enthalpy can be found by reading the enthalpy corresponding to the wet bulb at saturation. By adding the negative enthalpy deviation for the wet bulb and humidity ratio of the state point (from the small chart at left) the correct value of enthalpy can be obtained. Alternatively, the value of enthalpy for the given wet bulb at zero humidity can be read at the bottom of the main chart and corrected by adding the enthalpy of water shown on the small chart at left.

The chart is also essential in solving problems in manufacturing processes including the addition or

removal of moisture from the manufactured products.

Announced and discussed by Mr. Everetts, the chart was the climax of four papers on the subject of psychrometrics and the history and development of psychrometric charts. The first of these papers to appear in the JOURNAL will be in the February issue.

During the first part of the program, the chairman and vice chairman were, respectively, Eugene Ambrose of the American Electric Power Service Corporation, and Leo J. Riconda, General Mills Co. Professor Ralph G. Nevins, a member of both ASHRAE and ASME and Head of the Mechanical Engineering Department of Kansas State University, spoke on "Psychrometrics and Modern Comfort." His paper briefly traced the historical development of the ASHRAE comfort chart, including results of recent comfort research. He also discussed the proposed ASHRAE environmental research program to be carried out by Kansas State University.

Professor Ralph G. Nevins of Kansas State University speaks on Psychrometrics and Modern Comfort at the ASHRAE-ASME session on Wednesday morning, November 28

Dr. Joseph Lee Hollander, Associate Professor of Medicine and Chief of the Arthritis Section, School of Medicine, University of Pennsylvania, delivered a paper on the "Effect of Psychrometrics on Human Disease: A Study Utilizing the Controlled Climate Chamber." Dr. Hollander revealed that studies have been carried out on effects of variation of temperature, humidity, barometric pressure and atmospheric ionization in the controlled climate chamber in use at the University of Pennsylvania's arthritic rehabilitation center. This chamber was designed by the consulting engineering firm of Charles S. Leopold, Inc. In the chamber, two patients at a time have been confined continuously for periods of three to four weeks.

On the evening of the 28th, Dr. Hollander and Mr. Everetts repeated this talk at the meeting of the Div of Oceanography and Meteorology of the New York Academy of Sciences.

Dr. James Stokes of the Physiology Section of E. I. duPont de Nemours & Co., Inc., Haskell Laboratories, presented a paper prepared by the Chief of that Section, Dr. Lucien Brouha. Entitled "Physiological Reactions to Psychrometric Extremes," this paper discussed the

Dr. Joseph L. Hollander of the University of Pennsylvania had as his topic the Effects of Psychrometrics on Human Disease

Symposium Chairman, P. B. Gordon, ASHRAE Presidential Member, introduces President Everetts, who presented the Society's new psychrometric chart

reactions to cold and heat, chiefly those changes in respiratory, circulatory and thermoregulating mechanisms. Acclimatization and ways to reduce stress of extreme temperatures also were covered.

The second part of the ASHRAE program was chairmaned by P. B. Gordon, Presidential Member of the Society and Vice President of Wolff & Munier, Inc. Serving as Vice Chairman was Royal S. Buchanan of the American Motors Corporation. A paper prepared by Daniel D. Wile, ASHRAE Presidential Member and Executive Vice President of Recold Corporation, was read by Professor C. F. Kayan of the Department of Mechanical Engineering, Columbia University. Termed "Psychrometric Charts in Review," the paper defined the psychrometric chart as a primary tool for the solution of air-conditioning problems. Although many improvements have accumulated over the years, the basic principles have been little changed since Dr. Carrier presented his original theory and chart in 1911.

50th ANNIVERSARY HONORED

At the luncheon following the morning's technical program, Mr. Everetts presented a scroll to Cloud Wampler, Chairman of the Board and Chief Executive Officer of the Carrier Corp. The scroll commemorated the 50th anniversary of Dr. Willis H. Carrier's publication of "Rational Psychrometric Formulae." In his acceptance speech on

"The Legacy of Willis Carrier," Mr. Wampler predicted a big new market for air conditioning. He listed the five expanding markets, which are expected to produce nearly $6 billion in sales during the next five years, as central installations for homes, industrial plants, hospitals, schools and apartments.

As an example of progress in air-conditioning technology, Mr. Wampler described a thermoelectric heating and cooling unit occupying less than a cubic foot of space, which Carrier will deliver to the Navy in the spring of 1962. He stated that two units of this size would supply enough cooling for the average three-bedroom, 1200-sq ft house.

The morning session on Wednesday, November 29, concentrated on air conditioning. Chairman was Carl T. Ashby, ASHRAE Member and Director of Research and Engineering, Norge Div, Borg Warner Corporation. Vice Chairman was Ralbern H. Murray, Secretary of the Industrial and Commercial Gas Sect., American Gas Association.

Papers presented were: "The Economic Application of Gas Turbines to Large-tonnage Air-conditioning Installations" by C. R. Apitz, Sales Engineer, Clark Brothers Co., Div of Dresser Industries; "Absorption Air Conditioning" by G. Owen Kuhen, General Sales Manager, Carrier Corp.; and "Application of Natural Gas Engines to Air Conditioning and Refrigeration" by Robert A. D'Amour, Manager of Manufacturer Sales, Waukesha Motor Co.

In the afternoon, a four-member panel session reviewed the progress of the past 5 or 6 yr in the application of low-temperature phenomena. Among the subjects were superconductivity, superconducting devices, refrigeration and instrumentation, insulation materials and design principles.

1767

No. 1767

Psychrometrics and Modern Comfort

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Kansas State Univ.

Member
ASHRAE
ASME

Man has been concerned with psychrometrics, the state of the atmosphere, since the beginning of time and his comfort has been a prime consideration in his various activities. Maintaining a comfortable air temperature always has been a problem and many different methods have been proposed and used to achieve this end. Ventilation, the removal of heat, humidity and odors, was given engineering consideration as early as 1845.[1] The well publicized story of the Black Hole of Calcutta was a dramatic example of the effects of bad ventilation and over-heating.

In 1883, a German researcher, J. T. F. Hermans, unequivocally suggested that discomfort, commonly blamed on bad ventilation, in reality was due to excessive temperature and humidity.[2] Experimental evidence to support Hermans came in 1905 from work by Flugge and his associates.[3] Flugge concluded that: "the thermal properties of our atmospheric environment – temperature, moisture, air movement – are of far greater significance for our well being than the chemical properties of the air. The feelings of freshness which we experience when a closed room is freely ventilated or when we emerge into the outer air, are clearly due to more effective cooling of the body." German, English and American confirmation of Flugge's results led to complete acceptance of the thermal view of ventilation by 1909.

The sensation of comfort or the lack of awareness of discomfort has been found to be a complex subjective reaction resulting from a combination of physical, physiological and psychological

factors. Published research work shows that there are 15 or more factors which may affect the comfort sensation of a human being. Some of these factors are given in the table below.

Thermal environment	Age
Air temperature	Clothing
Air motion	Color
Relative humidity	Health
Mean radiant temperature	Noise
Physical activity	Lighting
Mental activity and attitude	Odors
Climate and season	Air ion content
Sex	Air pressure
Degree of adaptation	

Some of these factors, such as air temperature, influence our comfort to a truly large extent while others affect comfort but slightly.

The problem of defining ideal comfort also is complicated by the fact that an individual's reactions differ from day to day as much as from one to another so that a precise formulation of the 15 or more factors into requirements for ideal comfort is impossible. However, it is possible to establish limits which will satisfy the majority.

The principal factors which have the greatest influence on comfort are air temperature, air motion, mean radiant temperature and relative humidity. These factors make up the thermal environment. One other factor, air quality, is also important in many applications. The quality of the air must be such that odors, dust and general air pollution are reduced to a minimum or entirely eliminated.

Efforts to establish or specify conditions which would provide a comfortable environment began during the period 1913 to 1923. During this period, Professor John Sheppard at Teachers' Normal College in Chicago is reported to have introduced the term "comfort zone"[4]; The New York State Commission on Ventilation was appointed and began their numerous experiments, which resulted in a report published in 1923[5]; and the American Society of Heating and Ventilating Engineers published the first work of Houghten and Yaglon, which established "lines of equal comfort," defined "effective temperature" and determined the "comfort zone."[6]

These latter experiments were conducted under dynamic conditions with the subjects walking from one controlled room (temperature and humidity) to another. The conditions in the second room were adjusted until the relatively instantaneous reaction of the subjects gave identical comfort sensations or "equal warmth." These results were plotted on a psychrometric chart and were known first as lines of equal warmth. The effective temperature is defined as "an arbitrary index" which combines into a single value the effect of dry bulb temperature, humidity and air motion on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.[7]

The "comfort zone" and a "comfort line" were determined from a group of tests involving 130 subjects of both sexes, wearing different types of clothing and

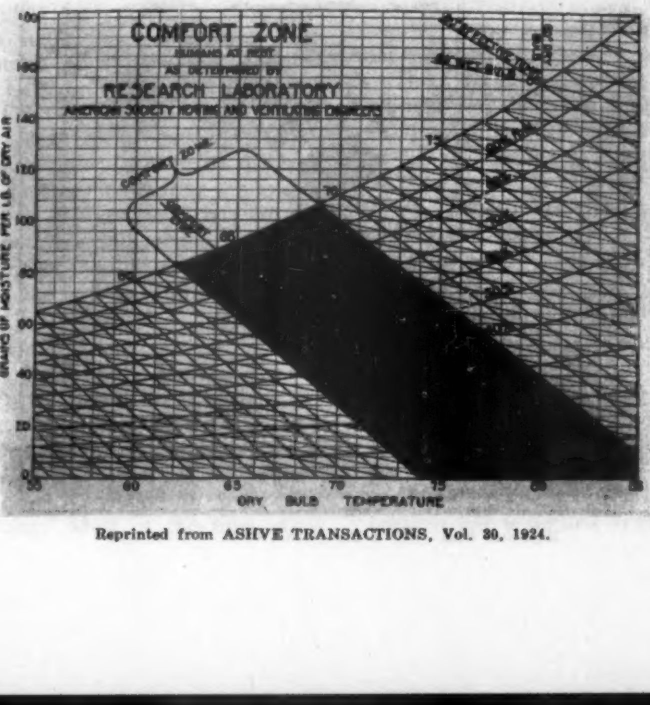
representing several different occupations. The subjects were seated in comfortable chairs and engaged in light activities such as card playing, reading and writing. Subjective reactions to the environments were obtained by asking

1. Is this condition comfortable or uncomfortable?
2. Do you desire a change?
3. If so, do you prefer warmer or cooler?

Twelve subjects were exposed to the test conditions for 3 hr, 14 subjects for 2 hr and 100 subjects for 15 min or somewhat longer. The comfort zone was defined as including those effective temperatures over which 50% of the people are comfortable. On this basis the zone limits were found to be 62 and 69 F ET with a comfort line of 64 F ET. Fig. 1 is a reproduction of the first comfort chart as it appeared in the introductory material of the Transactions of ASHVE published in 1924.

The papers of Houghten and Yaglou in 1923 were the first real attempt to relate comfort to temperature and humidity.[8] To extend the study an experiment was undertaken to determine the effect of air motion on comfort sensations. Three subjects stripped to the waist and wearing light-weight trousers were exposed to various environments in two test chambers in a manner similar to that described above except that one chamber

Fig. 1 Original ASHVE Comfort Chart



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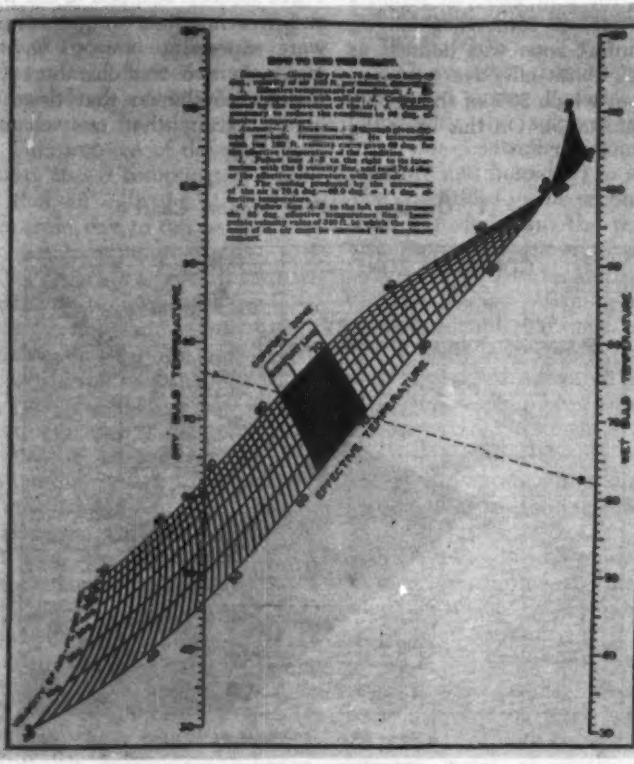
contained a bank of 15-in. fans to produce velocities across the subjects up to 500 fpm. When a condition of equal warmth was reported by the judges, the difference between the effective temperature in the still room and the effective temperature in the velocity room represented the cooling effect.

Obviously, the use of subjects stripped to the waist did not provide information needed by the engineer, so the next investigation studied the effects of clothing on comfort.[9] The clothing consisted of light-weight cotton underwear, madras shirt with collar attached, three-piece medium-weight woolen suit of medium-size mesh, cotton socks and shoes. The subjects were slightly active, having to walk be-

Fig. 2 Effective Temperature Chart showing the effects of air motion



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tween test chambers, manipulate the controls and take data. Also included in these tests were conditions of air motion involving the same velocities as before: 150, 300 and 500 fpm. From these data, the comfort zone was modified with limits of 63 and 71 F ET and a comfort line of 66 F ET. In the conclusion to this paper, the air conditioning engineer is admonished to use his judgment to shift the comfort line according to seasonal variations in clothing worn.

The results of this study were combined on a single chart, Fig. 2, relating dry bulb temperature, wet bulb temperature and air motion. Included on the chart were effective temperature lines and the comfort zone. This same chart, without the designated comfort zone, appears in the 1961 GUIDE AND DATA BOOK.

The collection of physiological data needed for a basic understanding of the man-environment process was not neglected by the ASHVE workers. In 1929, Houghten, Teague, Miller and Yant[10] published results showing the heat and moisture losses from the human body as a function of the effective temperature and air motion.

Further modification of the comfort chart resulted from work, published in 1929, by Yaglou and Drinker.[11] Their experiments were carried out at the Harvard School of Public Health to determine effects of summer climate on the comfort zone. Some 91 subjects were used (56 men and 35 women) with no restriction in regard to clothing. Most of the men wore two-piece palm beach or light woolen suits. The women wore silk, linen or cotton dresses. The subjective reactions were obtained using a 5-point scale: (1) cold, (2) comfortably cool, (3) very comfortable, (4) comfortably warm, (5) too warm. Test periods of approximately 3 hr were used with a pretest time of approximately one hr.

The summer comfort zone was found to be between 64 and 79 F ET. This zone included all votes indicating comfort, not just 50% or more. On this same basis the winter zone was found to lie between 60 and 74 F ET. The comfort lines were respectively 71 and 66 F. It is interesting to note that English people were reported at that time to prefer temperatures 8 F lower than those preferred by Americans. Also contained in this paper was considerable discussion of the relation of climate and season to the type of underclothing worn by Americans.

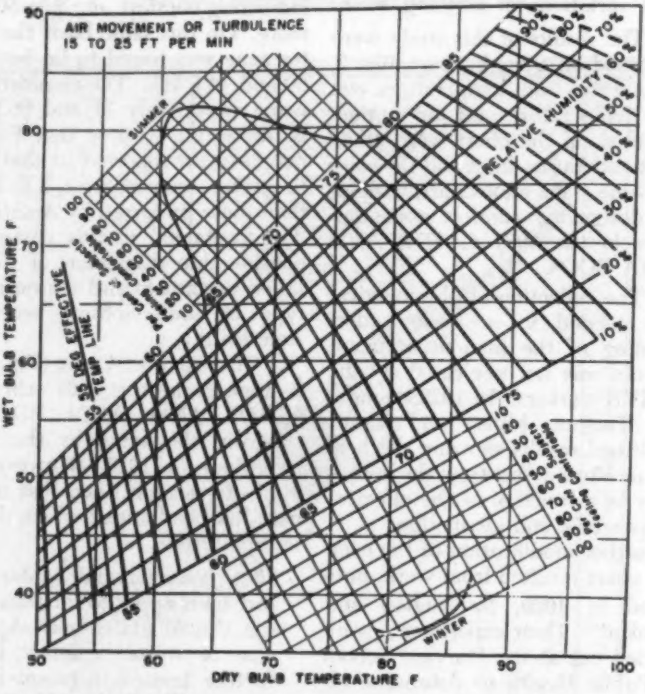
The Comfort Chart (Fig. 3) as it appears in the 1961 ASHRAE GUIDE AND DATA BOOK is basically the same as the chart published in 1925. However, in light of research since that time a note has been added which defines its application.

"Both summer and winter comfort lines apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes,

offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hr. The summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and Southern Canada, and at elevations not in excess of 1000 ft above sea level. An increase of one deg ET should be made approximately per 5-deg reduction in north latitude."

The areas indicating summer and winter comfort zones have been removed in light of field experience, careful analysis of the origi-

Fig. 3 ASHRAE Comfort Chart for still air



*Note—Both summer and winter comfort lines apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices, and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours. The summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and Southern Canada, and at elevations not in excess of 1000 ft above sea level. An increase of one deg ET should be made approximately per 5 deg reduction in north latitude.

†Dotted portion of winter comfort line was extrapolated beyond test data.

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nal data and laboratory experiments since 1940. Also the winter comfort data from the research prior to 1932 were eliminated and the data of Houghten, Gunst and Suciu[12] have been substituted. These data indicate an optimum condition of 68 ET in place of the 66 ET reported earlier. However, these data cover a range of effective temperature of but 65 to 69 ET.

Examination of the current chart indicates that along a constant effective temperature line an increase in relative humidity of 20% requires a decrease in dry bulb temperature of approximately 2 F. (In the extreme a change from 30% RH to 70% RH at 68 ET requires a change in dry bulb from 75 to 70.5 F.) Since 1938, evidence from various sources[10,14,15] has indicated that in the zone of thermal neutrality,* the effective temperature index over-emphasizes the effect of relative humidity on comfort.

In the range of dry bulb temperature of 73 to 77 F, variations in relative humidity from 25 to 60% do not affect comfort sensations, for sedentary or slightly active healthy men and women normally clothed in uniform environments.[16] Research by Glickman and others[17] using 15 subjects exposed to two levels of relative humidity (30 and 80% RH) found that the effective temperature index appeared to be adequate for subjects in the dynamic state but for equilibrium conditions the effective temperature index placed too much emphasis on relative humidity. Measurements in these tests included physiological as well as subjective reactions.

In 1947, Yaglou, one of the original investigators, proposed that the over-emphasis resulted from the use of instantaneous thermal impressions and the resulting adsorption and desorption phenomena, that is, the heat of adsorption giving a sense of warmth as moisture was adsorbed on skin and clothing.[18] Likewise, a cooling effect when the moisture evaporated. To correct the effective temperature index it was proposed that lines of constant mean skin temperature replace the ET lines. Yaglou's measured skin and clothing temperatures for men and women under ordinary summer and winter conditions are shown in Fig. 4. Lines of constant mean skin temperature in relation to effective temperature are shown in Fig. 5. Subjects normally dressed for a given dry bulb temperature in a range of 30 to 82 F were found to be comfortable when their mean skin temperature averaged between 91 and 93 F, provided subjects were not sweating. Skin temperature as a comfort index fails when subjects are more than slightly active or are sweating.

As early as 1950, the ASHVE (now ASHRAE) began planning a comprehensive program to reevaluate the comfort chart. A new environmental research facility was planned and built at the Cleveland laboratory. The results of the first

* Zone of vaso-motor regulation: heat production equal to the net heat loss by convection, radiation and evaporation with no change in stored energy and without sweating or shivering.

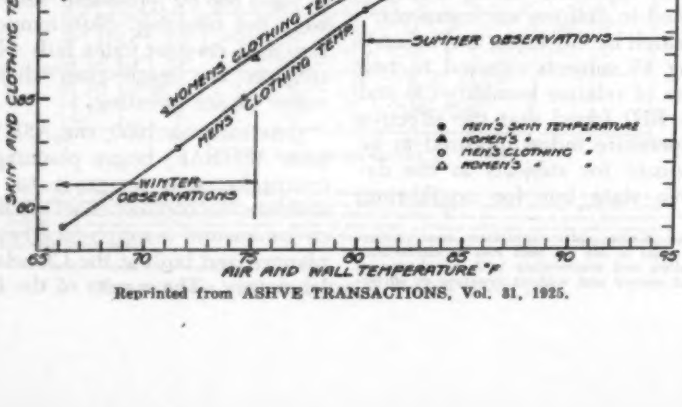
study (not actively initiated until 1957) were published in 1959 by Jennings and Givoni[19] and dealt with environment reactions in the 80 to 105 F zone, outside the comfort zone.

The second study, by Koch, Jennings and Humphreys, was published in 1960.[20] Their results can be considered the first step toward re-evaluation of the comfort chart. These tests were conducted with dry bulb temperatures from 68 to 94 F and with relative humidities from 20 to 90%. Twenty subjects, wearing summer clothing, were exposed for periods of three hr and were seated at rest. The data, shown in Fig. 5, show that over the range of variables studied the effect of relative humidity is small and that the optimum comfort line for these subjects was 77.6 F at 30% RH and 76.5 F at 85% RH.

The ideal comfort chart should give the air conditioning engineer as well as the environmental physiologist reliable data concerning the relation of psychrometrics and comfort. The dry bulb temperature and relative humidity are the basic factors which define our thermal environment. However, correction tables or charts are needed to show the effect of clothing, thermal radiation, air motion and activity on comfort sensation.

Clothing effects have been noted in many of the published reports. Differences which exist between the 1923 and the 1960 optimum comfort line probably are

Fig. 4 Mean skin and clothing temperatures in relation to environmental temperature



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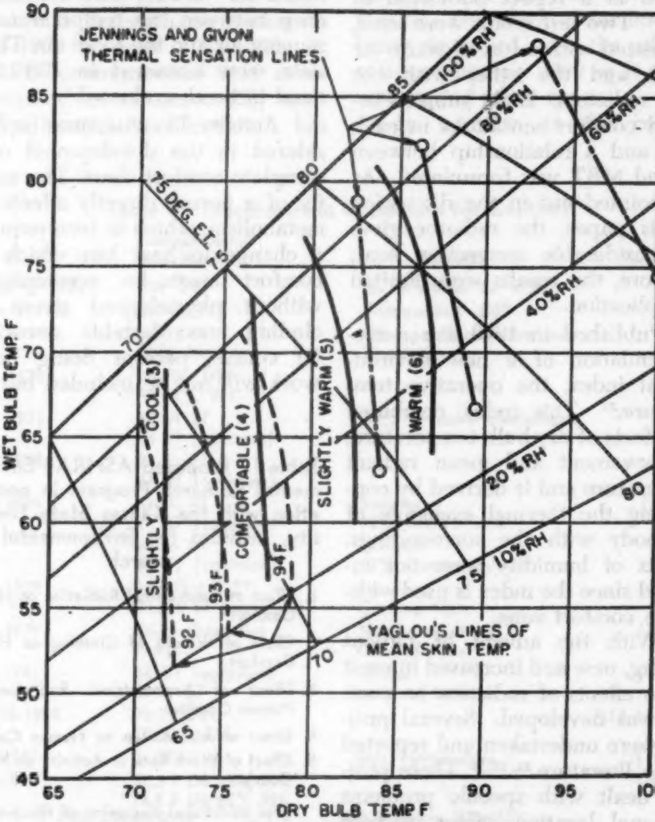
attributable to the change in clothing habits of Americans. Likewise, the difference in preferred indoor conditions which exist between the Americans and the British or Europeans usually is explained on the basis of differences in clothing.

The consideration of thermal radiation as a factor in comfort studies has also been of some concern from the beginning of comfort research. In a discussion to the 1923 paper[6] which first determined a comfort zone, J. G. Shodron called attention to the thermal radiation exchange between an oc-

Fig. 5 Thermal sensation lines, lines of constant mean skin temperature and effective temperature lines



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cupant and the enclosure which is necessary to produce comfort or a balance of heat production and heat loss. Other investigators have also called attention to this limitation in the applicability of the comfort chart.

The cooling effect produced by three cold walls of equal temperature was reported by Houghten and McDermott in 1933. Thermal radiation effects also were considered in a report published in 1941.[13] Two test rooms were used, one heated with a forced warm air system and the other with hot water radiators. Eight subjects reported comfort sensations in each room and a relationship between ET and MRT was formulated. As was pointed out in the discussion to this paper, the radiator gives off considerable convective heat; therefore, the results were limited in application.

Published in 1940 was a recommendation of a new environmental index, the operative temperature.[21] This index combined the effects of dry bulb temperature, air movement and mean radiant temperature and is derived by considering the thermal exchange of the body with the surroundings. Effects of humidity were not included since the index is used within the comfort zone.

With the advent of radiant heating, new and increased interest in the effects of radiation on comfort was developed. Several projects were undertaken and reported in the literature.[22,23,24] These projects dealt with specific problems of panel location, effect of floor surface temperatures, etc., and the results were such that a comfort zone with MRT as a variable could not be defined.

Air motion has always been considered an important variable and, as has been discussed before, some data are available. Criticism of a modern air conditioning system often can be traced to objectionable drafts. Research by Houghten[25] showed a relation between air velocity and the difference between the temperature of moving air and the room air. These data were obtained in 1938 and need to be re-evaluated.

Activity, likewise, must be considered in the development of a complete comfort chart. The activity of a person directly affects his metabolism, which in turn requires a change in heat loss which for comfort must be accomplished without physiological stress (including uncomfortable sweating). Of course, persons doing heavy work will not be included but the

Table I Proposed ASHRAE Environmental Research Program in cooperation with the Kansas State University, Institute for Environmental Research

1. Effect of Symmetrical Radiation on Human Comfort
2. Effect of Weight of Clothing on Human Comfort
3. Effect of Asymmetrical Radiation on Human Comfort
4. Effect of Air Motion on Human Comfort
5. Effect of Work Rate or Activity on Human Comfort

(For additional discussion of this program consult reference 26)

range of activity from resting to moderate work and dancing must be covered. Research has provided some information but much more is needed.

Future ASHRAE environmental research will attempt to provide the "complete" comfort chart. The proposed program is outlined in Table I.

This program will be carried out in cooperation with the Kansas State University Institute for Environmental Research, beginning in late 1962 or early 1963. The 12 x 24-ft environmental test room (described in reference 20), originally located in Cleveland, has been transferred to Kansas State University and will be housed in new quarters to be provided for the Environmental Research Institute. The staff of the Mechanical Engineering Department will conduct the research with the assistance of other University staff (Physician, Statistician, Psychologist, etc.). This project will expand the current environmental research program of the department sponsored by The National Institute of Health. The physical facilities are being expanded with the addition of approximately 3500 sq ft of laboratory and office space.

In addition to the comfort

Table II Psychrometric Conditions for Comfort

Date	Environmental Specifications	Remarks	Reference
Prior to 1900	70F DBT	no reference to W.B.T.	(6)
Early 1900	56F WBT	disregarding DBT	(6)
1914	68F DBT 40% RH	Chicago Commission on Ventilation	(5)
1923	66-72F DBT 19-61% RH		(5)
1923	62-69 ET 64 ET (optimum)	50% or more comfortable	(6)
1925	63-71 ET 66 ET (optimum)		(9)
1929	66-75 ET 71 ET (optimum) 60-74 ET 66 ET (optimum)	Summer 98% Comfortable Winter 98% Comfortable	(11)
1939	64.8-76.0 F ET 71.8 F ET (optimum)	Basal Clothed Men	(29)
1939	74-84 Operative Temp.	Non-basal, at rest Clothed Men	(30)
1941	68 ET (optimum) 65-69 ET	Winter range of tests	(12)
1938-1956	73-77 DBT 25-60% RH		(13, 14, 15)
1947	91-93F	Skin Temperature	(18)
1960	77.6 F DB 30% RH 76.5 F DB 85% RH		(20)

DBT—Dry Bulb Temperature
WBT—Wet Bulb Temperature
RH—Relative Humidity
ET—Effective Temperature

aspects, ASHRAE is interested in the effects of the thermal environment on learning rate and achievement and on productivity. Attempts will be made to collect and disseminate the results of research in this area, whether sponsored by ASHRAE or by some other agency.

The development of the psychrometric specifications for the comfort zone and optimum comfort line is summarized in Table II. It is interesting to note the shift in the optimum conditions since 1900 and the reluctance of some researchers to define a line, preferring to specify a range or zone of comfort. Leopold, in his paper "Conditions for Comfort"[27] proposed the use of a discomfort index rather than a comfort index. This procedure was also used by Chrenko in England.[28] An argument for accurate control of environmental conditions was developed by Leopold in the same paper, showing that even though persons may individually have a wide zone of comfort, it is necessary to maintain close control of the environment for maximum group comfort.

The engineer must seek to provide environmental conditions which will meet effectively the comfort requirements of a heterogeneous population. There must be a compromise such that the maximum number of people is comfortable. Since each person is comfortable within a range of temperatures, the engineer need not provide a fixed optimum temperature for each individual.

As pointed out by Bedford,[29] the establishment of the proper effective temperature does not ensure that the environment will be pleasant. Factors such as air movement, thermal gradients, and radiant effects should be considered. Herrington, in a discussion to this paper, points out the need to consider also the effects on comfort response of age, sex, clothing and activity.

It is the objective of ASHRAE and the cooperating institutions to provide the information necessary for design of effective and efficient thermal environments in which comfort for the maximum number of occupants is achieved, and to provide basic physiological data relating man to his environment, thereby helping to increase our understanding of the human factor in the creation of a comfortable, healthful, pleasant and efficient environment.

REFERENCES

1. Bernan, Walter, History and Art of Warming and Ventilating Rooms and Buildings, London, 1845.
2. Hermans, J. T. F., "Veber die vermeintliche Ausathmung organischer Substanzen durch den Menschen", Archiv fur Hygiene, i, 1, 183.
3. Flugge, C., "Veber Luftverunreinigung, Wormestauung and Luftung in geschlossenen Raumen" Zeitschrift fur Hygiene, XLIX, 363, 1905; Heymann, p. 388; Paul, p. 405; Ercklentz, p. 433.
4. Hill, E. Vernon, Discussion to "Determination of the Comfort Zone" by F. C. Houghten and C. P. Yaglou, Trans. ASHVE, v. 29, 1923.
5. Report of the New York State Commission on Ventilation, E. P. Dutton and Co., New York, 1923.
6. Houghten, F. C. and Yaglou, C. P., "Determining lines of Equal Comfort" and "Determination of the Comfort Zone", Trans. ASHVE, v. 29, 1923.
7. ASHRAE GUIDE AND DATA BOOK 1961, Fundamentals and Equipment, ASHRAE, 1961.
8. Houghten, F. C. and Yaglou, C. P., "Cooling Effect on Human Beings Produced by Various Air Velocities", Trans. ASHVE, v. 30, 1924.
9. Yaglou, C. P. and Miller, W. E., "Effective Temperature with Clothing", Trans. ASHVE, v. 31, 1925.
10. Houghten, F. C., Teague, W. W., Miller,

W. E., Yant, W. P., "Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems", Trans. ASHVE, v. 35, 1929.
11. Yaglou, C. P. and Drinker, P., "The Summer Comfort Zone: Climate and Clothing", Trans. ASHVE, v. 35, 1929.
12. Houghten, F. C., Gunst, S. B. and Suciu, J., "Radiation as a Factor in Sensation of Warmth", Trans. ASHVE, v. 47, 1941.
13. Winslow, C. E. A., Herrington, L. P., and Gagge, A. P., "The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity", Am. Jour. Physiol., 1938.
14. Newburgh, L. H., Physiology of Heat Regulation, W. B. Saunders, Philadelphia, 1948.
15. Rowley, F. B., Jordan, R. C., and Snyder, W. E., "Comfort Reactions of 275 Workers During Occupancy of Air Conditioned Offices", Trans. ASHVE, v. 53, 1947.
16. Fahnestock, M. K. and Werden, J. E., "Environment, Comfort, Health and People", Refrigerating Engineering Feb., 1956.
17. Glickman, N., Inouye, T., Keeton, R. W., Fahnestock, M. K., "Physiological Examination of the Effective Temperature Index", Trans. ASHVE, v. 56, 1950.
18. Yaglou, C. P., "A Method for Improving the Effective Temperature Index" Trans. ASHVE, v. 53, 1947.
19. Jennings, B. H. and Givoni, B., "Environment Reactions in the 80 to 105 F Zone", Trans. ASHRAE, v. 65, 1959.
20. Koch, W., Jennings, B. H., and Humpheys, C. M., "Sensation Responses to Temperature and Humidity Under Still Air Conditions in the Comfort Range", Trans. ASHRAE, v. 66, 1960.
21. Gagge, A. P., "Standard Operative Temperature", Am. Jour. Physiol, vol. 131, 1940.
22. Herrington, L. P. and Lorenzi, R. J., "Effect of Panel Location on Skin and Clothing Surface Temperature", Trans. ASHVE, v. 56, 1950.
23. Shrenko, F. A., "Heated Floors and Comfort", Jour. Institution of Heat. and Vent. Engrs. (London) v. 23, January, 1956.
24. Nevins, R. G. and Flinner, A. O., "Effect of Heated Floor Temperatures on Comfort", Trans. ASHAE, v. 64, 1958.
25. Houghten, F. C., Gutberlet, C. and Witkowski, E., "Draft Temperatures and Velocities in Relation to Skin Temperatures and Feeling of Warmth", Trans. ASHVE, v. 44, 1938.
26. Nevins, R. G. and Humphreys, C. M., "Proposed Environmental Studies in the ASHRAE Research Program", ASHRAE JOURNAL, Jan. 1961.
27. Leopold, C. S., "Conditions for Comfort", Trans. ASHVE, v. 53, 1947.
28. Bedford, T., "Research on Heating and Ventilation in Relation to Human Comfort", Trans. ASHRAE, v. 65, 1959 (Discussion by L. P. Herrington).
29. DuBois, E. F., "Heat Loss from the Body", Bull. New York Academy of Medicine, 2nd Series, v. 15, 1939.
30. Gagge, A. P., Winslow, C. E. A. and Herrington, L. P., "The Influence of Clothing on the Physiological Reactions of the Human Body to Varying Environmental Temperatures", Am. Jour. Physiol. v. 124, 1938.

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No. 1768

Physiological Reactions to Psychrometric Extremes

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Psychrometric extremes imply both cold and warm conditions and we shall discuss successively the physiological reactions of man exposed to these environments. From a practical point of view, the problem is to find how man can adjust to extreme conditions of temperature, humidity and wind velocity without impairing his health and with a working capacity which, although reduced, is still satisfactory.

COLD

Man is in a more favorable position to counteract the effects of low temperatures than those of high temperatures. Heat loss from the body exposed to cold is increased through radiation, conduction and convection. In body insulation, two thermal gradients have to be considered: a – the temperature drop between deep body temperature and that of the skin; b – the temperature difference between the skin surface and the environment. By using suitable clothing, protection against excessive heat loss can be achieved down to extremely low temperatures.

The fact that in the arctic the caloric intake is but slightly larger than in moderate climates indicates that man does not expose himself to cold sufficiently to increase his average heat loss appreciably more than in moderate environments. Nevertheless, heat loss can occur and when it reaches a certain level, vasoconstriction and then shivering take place—the former reducing heat loss through the skin, the latter increasing body heat production as a defense mechanism against cold.

Physical activity has the same effect but is not as efficient. For example, Carlson et al.[1] have shown that at 50 F exercising at a rate more than three times the efficient shivering metabolic level does not prevent body cooling in a man wearing shorts because of the large convective heat loss incurred by

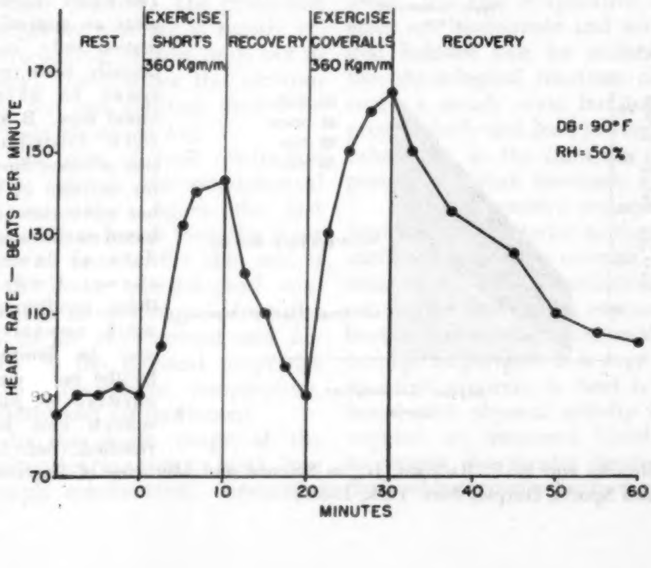
the unprotected legs during pedaling.

Furthermore, physical activity increases the blood flow in the limbs and thus aids in the loss of heat. Even light work will produce that effect and, for a man lightly clad, exercise is not a good source of heat for maintaining body temperature. Shivering is more efficient because heat loss is minimized by reducing the exposed body surface by "curling up" and also maintaining the insulation by vasoconstriction.

Relative humidity of the air has little effect on cooling rates during cold exposure but seems to affect the amount and severity of shivering (Burton, et al.[2]). On the other hand, the clothing insulation is reduced due to dampness and the body cooling rate can be 50% greater than that of subjects in dry clothing. This is what occurs most frequently when gloves and shoes become wet due to sweating during physical exercise in the cold.

The effect of air movement in the cold is to lower the ambient temperature. Siple and Passell[3] have introduced a "wind chill index" to measure the cooling power of the wind, different combinations of temperature and wind velocity having the same physiological effect. For example, exposed flesh will freeze if the wind velocity is 45 mph and the air temperature 20 F, and the same effect will be

Fig. 1 Additional cardiovascular stress produced by light clothing at 90 F and 50% R.H.



produced by a temperature of –40 F and a wind velocity of but 2 mph.

Froese and Burton[4] have found that the tissue insulation of the fingers exposed to cold may increase sixfold, whereas that of the head does not. Thus, there is a practical need for keeping the head warmly covered in order to prolong endurance time in the cold. In spite of the large increase in the insulation of the fingers, a tight vasoconstriction of the extremities to prevent heat exchange is not desirable since the result is a progressive loss of function of the hands and feet and finally tissue injury.

Cardiovascular reactions other than vasoconstriction are seen often during cold exposure: greater frequency of changes in electrocardiograms after light work, in-

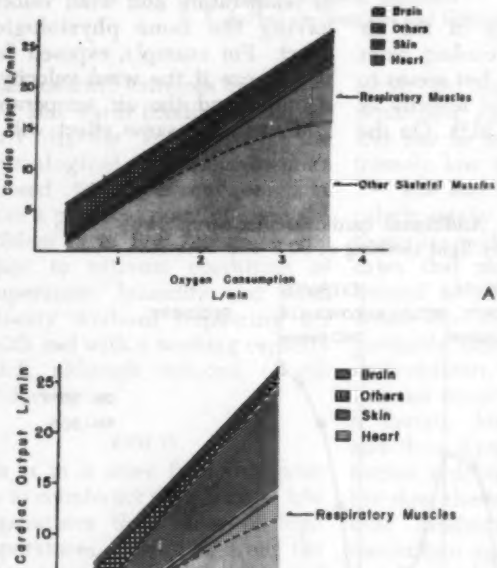


Fig. 2 Distribution of total cardiac output to various areas as a function of oxygen consumption; values reached after short periods of exercise at normal temperature. A at 70 F. The figure represents an approximation only, especially with respect to skin blood flow. B at 100 F. Distribution of blood flow for exercise in a hot environment, based on the same data as lower curve. Under these conditions, work capacity may be limited by the maximum cardiac output which can be reached. (Ref.: L. Brouha and E. P. Radford, Jr., in Science and Medicine of Exercise and Sports, Harper, New York, 1960.)

crease in arterial and venous blood pressure with or without changes in heart rate. Whether or not these higher blood pressures increase the circulation and therefore decrease the insulation of the body is not known.

The conclusion is that man can adapt himself quite successfully to the ambient conditions at the cold end of the psychrometric chart mostly through clothing and, when need be, by increasing his metabolic heat production. Under these conditions, he carries with him his own "microclimate" which is more or less comfortable but, in any case, much warmer than the outside environment.

HEAT

As opposed to cold exposure, it is only within rather narrow limits that man can protect himself against warm surroundings by adequate clothing. The protection afforded by clothing is mostly effective against radiant heat; but in general, the warmer the environment the less clothing should be worn, as shown in Fig. 1.

The rate of heat dissipation mainly controls the physiological reactions. Here again, the first stage of heat loss is from the inner source of heat to the skin and is largely under physiological control. The second stage is from the skin to the environment and depends on the physical properties of the atmosphere: temperature, humidity and air movement.

In the warm range of the psychrometric chart, heat loss through conduction, convection and radiation is less than the heat produced by muscular activity or gained from the environment or both: cooling must take place by evaporation of sweat, and sweating increases progressively up to a maximum rate when ambient conditions and work load become more severe. Two situations occur: a – the evaporation of sweat is sufficient to cool the body and a thermal equilibrium is achieved with a constant internal temperature.

Under these conditions, pulmonary ventilation, oxygen consumption, blood pressure, cardiac output and heart rate increase at the beginning of muscular activity and reach a steady state, the level of which is determined by the work rate for a given environment. When work stops, recovery takes place and these functions return progressively toward their resting level. b – The evaporative processes are inadequate and no thermal balance can be maintained: the physiological reactions do not reach a steady state but increase progressively and lead toward heat exhaustion as the duration of exposure and work increases.

Since convective transport of heat from the interior to the body surface is one of the essential functions of the blood circulation, one can expect that during exposure to heat cardiovascular reactions should be most important. It is even more so when exposure to heat is combined with physical activity which requires an increased blood flow to supply the greater demand of the working muscles for fuel, mainly

oxygen. In an automobile, the excess heat of the engine is transferred to the outside air via a circulation of water and a radiator; the fuel is provided by gasoline using a separate system. In man, blood is called upon to transport fuel as well as heat and the requirements of these two functions are often in conflict since the total amount of blood available and the speed at which it can be circulated are limited.

Because the warm environment hinders heat loss, the cardiovascular system may reach its "maximum heat transfer" ability before it reaches its peak "energy transportation" ability. Therefore, any increase in work rate beyond this point can only be achieved at the expense of a rise in body temperature. Such rises are not dangerous over a small range, but body temperatures maintained above 102 F are tolerated poorly.

If work is to continue, the work rate must be cut down to a point where the cardiovascular system can again handle the resulting heat production. Thus, psychrometric extremes in the warm range impose upon the body a thermal burden which is composed of two parts: the first is the interference with heat loss from the body, the second is the heat which actually is gained by the body from the high

Table I Average Oxygen Consumption Variations in Males and Females Working in Three Environments

Oxygen in l/min

	MALES			FEMALES		
Phases*	Normal	Warm-dry	Warm-humid	Normal	Warm-dry	Warm-humid
I	0.330	0.370	0.390	0.260	0.290	0.270
II	1.470	1.270	1.420	1.180	0.980	1.070
III	1.490	1.390	1.550	1.200	1.020	1.120
IV	2.290	1.910	2.160	1.780	1.570	1.620
V	0.430	0.410	0.490	0.350	0.290	0.370
VI	0.320	0.360	0.340	0.280	0.220	0.260

* I = resting; II = first 5 min of pedaling at submaximal work rate; III = submaximal work; IV = maximal work; V = resting on bicycle; VI = resting in armchair.

Table II Average Heart Rate Variations in Males and Females Working in Three Environments

	MALES			FEMALES		
Phases*	Normal	Warm-dry	Warm-humid	Normal	Warm-dry	Warm-humid
I	81	89	89	86	101	99
II	128	133	132	138	140	145
III	144	152	156	151	164	170
IV	172	175	180	176	179	179
V	113	115	128	114	120	130
VI	86	91	99	93	100	103

* I = resting; II = first 5 minutes of pedaling at submaximal work rate; III = submaximal work; IV = maximal work; V = resting on bicycle; VI = resting in armchair.

ambient temperature. The second effect aggravates the first since it is adding heat to a system already impaired in losing it. Muscular activity increases the burden further.

Fig. 2 shows the effect of a high environmental temperature on the distribution of blood flow and illustrates the marked increase in the skin blood flow with increasing oxygen consumption; i.e., harder work producing more heat.

When, in spite of the need for an important skin blood flow, the work load remains at a high level, oxygen delivery to the muscles has priority and the work must be done at the expense of an increase in body temperature due to inadequate cooling. However, the time during which work can continue is limited, and most men will stop when their internal temperature exceeds 102 F.

Because of the importance of knowing the physiological reactions to different work loads at various high temperatures and relative humidities, experiments under strictly controlled conditions of work load and heat load have been pursued for the last eight years at Haskell Laboratory. Some of our results are summarized hereafter.

1. Effect of Continuous Work.[5] The subjects were six male and six female laboratory workers in good health. The work consisted in pedaling a bicycle ergometer at 60 rpm at a submaximal load for 30 min, immediately followed by 4 min at maximal load; recovery was followed for one hr. Three environmental conditions were used:

77 F and 42% R.H., effective temperature = 70.5 F or normal;
99 F and 25% R.H., effective temperature = 82.5 F or warm-dry;
90 F and 82% R.H., effective temperature = 86.0 F or warm-humid.

Heart rate, pulmonary ventilation, oxygen consumption and carbon dioxide production were recorded throughout the experiment. Blood pressure was measured every minute, oral temperature every 15 min and body weight at the beginning and at the end of the experiment. Tables I and II summarize the results for oxygen consumption and heart rate.

Oxygen consumption was approximately the same at normal temperature and under warm-humid conditions, whereas it was lower under warm-dry conditions. The same was true for pulmonary ventilation, indicating that in the warm-dry environment efficiency was increased slightly, a fact which is substantiated by concomittant lower levels of blood lactic acid.

On the other hand, heart rate increased when going from normal to warm-dry and even more to warm-humid. Oral temperature after exercise showed an average gain of 0.5 F in normal and in warm-dry conditions, but in the warm-humid surroundings the gain was 1.5 F for the men and 1.0 F for the women; thermal balance was achieved in the former case

but not in the latter. Weight loss was most significant in the warm-dry atmosphere where evaporation of sweat was greatest. It was less in the warm-humid environment where evaporation was reduced because of the high water vapor content of the air. Average blood pressure changes were the same under the three environmental conditions.

The conclusions from these experiments are as follows:

(a) The stress on the cardiovascular system increases with the ambient temperature. The greatest stress was observed when the work was performed in an environment combining high temperature and high relative humidity.

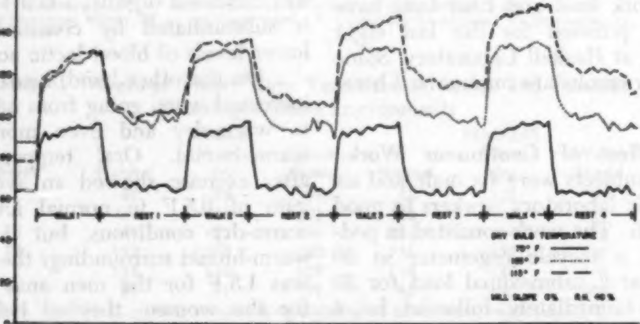
(b) Heart rate during work and recovery varies according to work load, ambient conditions and sex.

(c) In contrast, blood pressure varies with work load only and is not significantly influenced by sex or by environment.

(d) Pulmonary ventilation and oxygen consumption are determined by the work load but influenced little by the environment. Work under warm-dry conditions appears to be performed with an oxygen consumption slightly lower than in the other environments.

(e) Sweat loss and weight loss are greatest under warm-dry conditions; under warm-humid condi-

Fig. 3 Effect of repeated work-rest cycles. The work load was constant at 0 mill slope and the relative humidity constant at 40%. Dry bulb temperature was increased from 70 to 90 to 115 F. A steady state from cycle to cycle is maintained at 70 and 90 F, but at 115 F no steady state can be maintained and each cycle produces higher reactions during work and more incomplete recovery



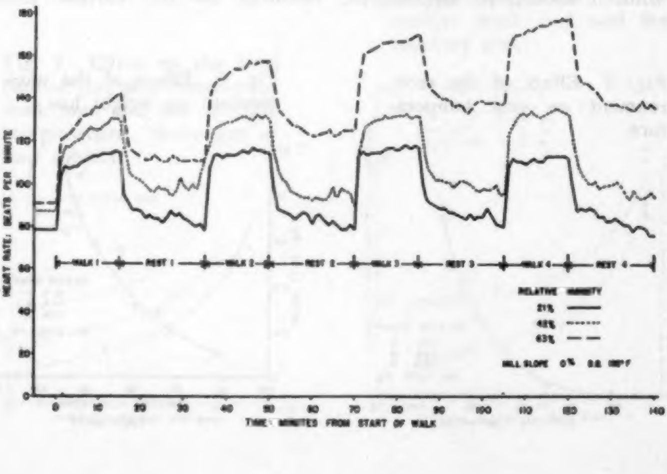
tions evaporative cooling is impaired and body temperature rises appreciably.

2. Effect of Repeated Work Cycles – In warm environments the reactions often are modified when, instead of working for one single period of time, the subjects perform several work periods with rest periods in between. To study this problem, two sets of experiments were made, one in which all work-rest cycles were of the same duration, another in which the duration varied.

Repetitive work cycles of constant duration.[6] The work was performed by four college students and consisted of walking on a treadmill at 3.2 mph for 15 min followed by a rest of 20 min. This cycle was repeated four times. Three treadmill slopes were used: 0, 5 and 10% grades. Environmental conditions varied from 70 to 120 F and 21 to 83% R.H. Wind velocity was approximately 50 fpm.

Fig. 3 shows the average heart rate changes that were observed when the dry bulb temperature was increased from 70 to 115 F, with a treadmill slope of zero and a R.H. of 40%. A steady state was achieved at the lower temperatures, but at 115 F a progressive increase occurred from cycle to cycle during work and recovery. Fig. 4 shows the same phenomenon when the slope of the treadmill was zero and the dry bulb temperature 105 F, but when the R.H.

Fig. 4 Same experiments as in Fig. 3. For a constant work load and a constant dry bulb at 105 F, the relative humidity was increased from 21 to 63%. A steady state from cycle to cycle is maintained at 21 and 43%, but at 63% each cycle produces higher reactions during work and recovery



was increased from 21 to 63%: higher heart rates were observed from cycle to cycle in the most humid environment.

The results demonstrate clearly that at 105 F and 63% R.H.; i.e., an effective temperature of 94.7 F, accumulative strain was marked. One of the subjects reached exhaustion and was able to complete but three cycles. This accumulative strain also was present at 115 F and 39% R.H. and at 120 F and 21% R.H.; i.e., effective temperature of 95 and 91.5 F, respectively.

When the work load was increased by walking on a 5% grade, the same phenomenon occurred at a lower effective temperature of 89 F. Experiments with a 10% grade could not be carried out in these hot environments. Our data do not permit defining exactly the dry bulb or the R.H. which initiates accumulative strain. A permissible estimate is that for light and moderate work, with an oxygen consumption of 0.8 to 1.5 liter per min, it should be anticipated at dry bulb temperatures above 90 F when the R.H. is high (80%) and above 105 F when the R.H. is low (20%). In terms of the basic scale of effective temperature, the critical region appears to be between 88 and 91 F.

Changes in body temperature and weight loss between the beginning and the end of the experiments are shown in Figs. 5 and 6, respectively; they increased markedly as the effective temperature rose.

In addition, the following data were calculated:

(a) **Total cardiac cost** above rest; i.e., the number of heart beats above the resting level that were needed to perform the work and to recover for 20 min.

(b) **Cardiac efficiency**, which is the amount of external work produced per extra heart beat and is expressed here as meters walked per beat (Brouha, et al.[5]).

The effect of work and environment on the cardiac cost is

Fig. 5 Effect of the environment on oral temperature

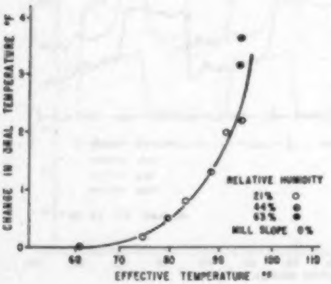
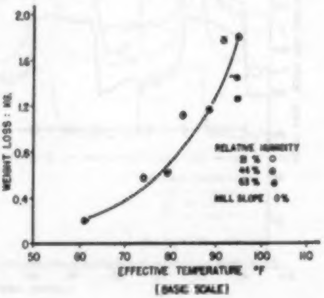


Fig. 6 Effect of the environment on weight loss

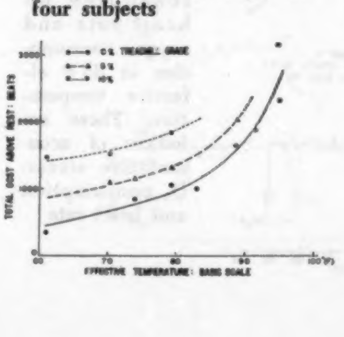


shown in Fig. 7 for the three grades of work. It is clear that for a constant work rate, the cardiac cost is greater when the environmental stress is more severe. Conversely, for a given environmental stress the cardiac cost increases with the work rate.

When the cardiac cost of work and the cardiac cost of recovery are plotted separately against effective temperature, it should be noticed that as the environmental stress increases, a larger proportion of the extra cost is due to the cost of recovery. In other words, for the same amount of work performed, the higher the effective temperature the more time it takes to recover (Fig. 8).

On the other hand, cardiac efficiency decreases as the environmental stress increases. This is shown in Fig. 9 where cardiac efficiency and oral temperature are plotted against effective temperature; the reciprocal relation between these two variables is evident.

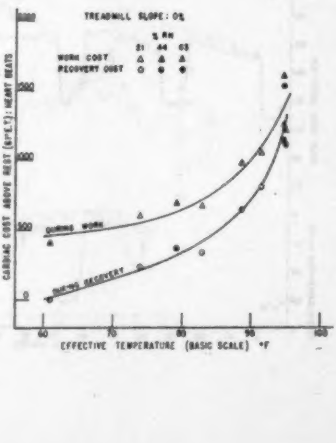
Fig. 7 Effect on the total cardiac cost of increasing the work load and the effective temperature. Averages of four subjects



Repetitive work cycles of different duration.[7] Similar studies were made in which the duration of the work-rest cycles was modified. In typical experiments the subjects performed 25 min of work followed by 10 min of rest repeated four times, and 50 min of work followed by 20 min of rest repeated twice. The total amount of work and of heat exposure was exactly the same with the two schedules. Fig. 10 gives the results observed for oxygen consumption and heart rate when the subjects were walking the treadmill on a 5% grade at 3.2 mph in an environment at 85 F effective temperature.

Regardless of the sequence used, oxygen consumption reached the same steady state and did not indicate any accumulative strain. Heart rate did likewise when the 25-min work – 10-min rest sched-

Fig. 8 Effect of increasing effective temperature on the cardiac work cost and the recovery cost



ule was performed. However, with the 50-min work – 20-min rest schedule, a definite increase in heart rate took place during each work period and was definitely greater during the second period than during the first, both at work and in recovery. These results emphasize the importance of determining adequate work-rest periods for men engaged in physical activity in warm surroundings, a situation frequently encountered in industrial operations.

In these warm environments and for a constant oxygen consumption, the pulmonary ventilation can increase more than it does in the cool. Fig. 11 compares the curves obtained during the first and the third cycles with the 25-min work – 10-min rest schedule. In this case, for virtually the same oxygen consumption, a definite increase in pulmonary ventilation from the first to the third cycle is evident. This observation has been made repeatedly at levels of energy

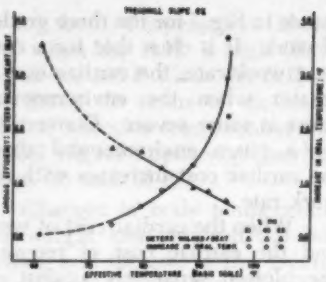


Fig. 9 Effect of effective temperature on oral temperature and cardiac efficiency

expenditure below 1.0 liter of oxygen per min and with an average body temperature increase of 0.5 F, provided the environmental stress is high.

This phenomenon is important in industries where the workers are exposed to toxic atmospheres, since heat inducing a greater hyperventilation will lead to a greater inhalation of dangerous substances. Fig. 11 also shows that, of all the

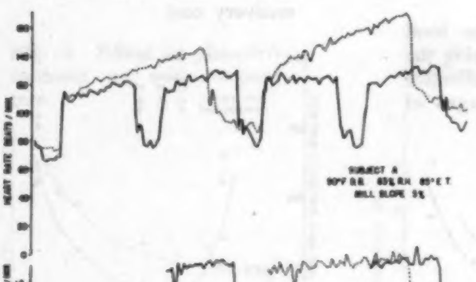
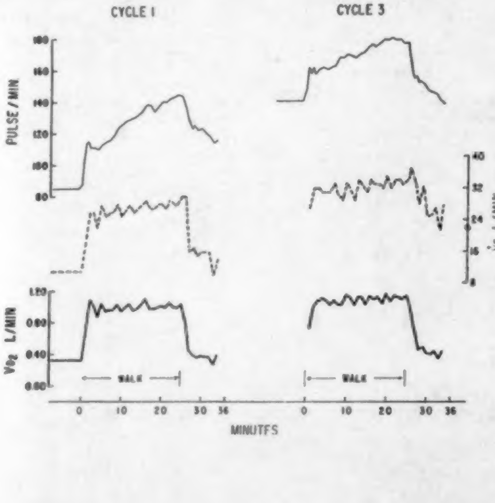


Fig. 10 Effect of two work-rest schedules on heart rate and oxygen consumption at 85 F effective temperature. These are indices of accumulative stress: O_2 consumption and heart rate

variables that we have studied in all these experiments, the changes in heart rate are the most sensitive measurements to detect the combined stress produced by the work and by the environment. Simple methods have been devised that are applicable easily in the plants and give a reliable evaluation of the total stress produced by any given job performed under psychrometric extremes in the warm range (Brouha[8]).

3. Effect of Clothing and Protective Equipment – Clothing and various kinds of protective equipment increase the stress of work and of the environment by hampering heat dissipation. Any garment or equipment that interferes with the evaporative cooling processes will put an additional load on the heart, sometimes to a considerable degree. Consequently, there is a close relation between the effect of clothing, the work load and the temperature of the environment. This is shown in Fig. 12 which summarizes the results of the effects of two grades of work in different environments with and without protective clothing against radioactive substances. The upper heart rate curves show that in a comfortable environment, 72 F and 64% R.H., and for the lighter load, 360 kg-m/min, wearing the protec-

Fig. 11 Relation between heart rate, oxygen consumption and pulmonary ventilation. Comparison of cycle I and cycle 3 for the 25-min work–10-min rest schedule. Dry bulb 105 F; R. H. 63%; effective temperature 94.7 F



tive outfit increased the cardiovascular reactions during work but slightly and did not interfere with the recovery processes. For the heavier load, 540 kg-m/min, the cardiovascular reactions were definitely higher both during work and during recovery, but still within reasonable limits.

The striking increase of this additional stress in a warmer environment, 115 F and 22% R.H., is illustrated by the lower curves. Here, wearing the protective outfit produced markedly higher reactions during work and a definite delay in the recovery processes even at the lighter work load. Intermediate results were obtained in less warm surroundings.

Similar effects are observed when impervious clothing is used for protection against chemicals. These garments interfere maximally with the dissipation of body heat. The layer of air trapped inside the suit warms up rapidly to body temperature and quickly becomes saturated with moisture. No body cooling by evaporation can take place, and the strain increases steadily toward high heart rates

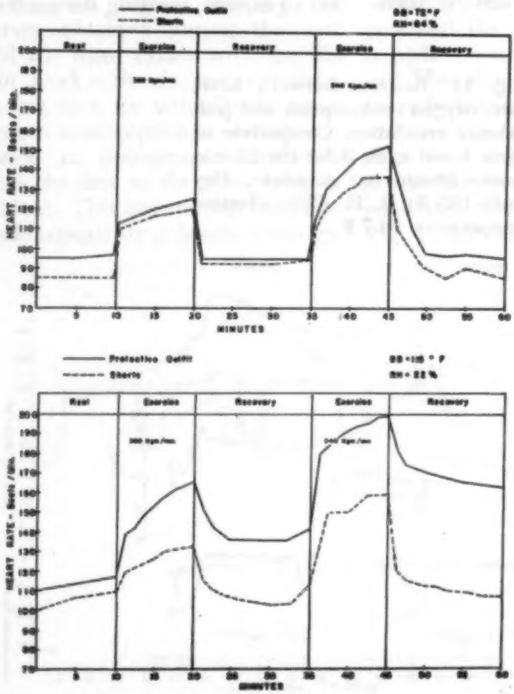


Fig. 12 Influence of temperature, work load and protective clothing on heart rate

and high body temperatures; such reactions are present even when the temperature of the environment is favorable and the work load light.

Additional stresses of varying degree occur when rubber boots, respirators, air masks, heavy aprons, etc., are worn. Clothing must, therefore, be taken into consideration in evaluating the physiological requirements of a job, especially in warm surroundings. Little attention has been given to the adverse physiological effects of these factors and more research is needed to assess accurately the importance of "clothing stress" in industrial operations.

REFERENCES

1. Carlson, L. D., D. Pearl and W. Scheyer, Am. J. Physiol. 179, 625, 1954.
2. Burton, A. C., R. A. Snyder and W. G. Leach, J Appl. Physiol. 8, 269, 1955.
3. Siple, P. A. and C. F. Passell, Proc. Am. Phil. Soc. 89, 177, 1945.
4. Froese, G. and A. C. Burton, J. Appl. Physiol. 10, 235, 1958.
5. Brouha, L., P. E. Smith, Jr., R. Delanne and M. E. Maxfield, J. Appl. Physiol. 16, 133, 1960
6. Brouha, L. and M. E. Maxfield. To be published.
7. Brouha, L., M. E. Maxfield, G. J. Stopps and P. E. Smith, Jr. To be published.
8. Brouha, L., Physiology in Industry, Pergamon Press, London, 1960.

No. 1769

Effect of Climatic Factors on Human Disease

JOSEPH LEE HOLLANDER, M.D.
Associate Professor of Medicine,
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Although medical annals contain many reports indicating that weather conditions affect many human disease states, facilities for complete control of all climatic factors for continuous observation on human patients have been lacking. Utilizing a universal Controlled Climate Chamber, in which two patients at a time have been confined continuously for periods of 3 to 4 weeks, individual variations in temperature, humidity, barometric pressure and atmospheric ionization have been studied on twelve arthritic patients. Preliminary results, not yet conclusive, indicate that no single factor is responsible for the "weather effect" experienced by these individuals.

Shelves of medical libraries are filled with volumes dealing with the effects of climate on human disease, but there is a dearth of scientific observation under completely controlled conditions. Empiricism characterizes a tremendous proportion of the articles, some extolling the value of salt air, others, the mountains or the desert. Observations largely consist of two types: diaries of individual patients who made day-to-day observations on the weather and how they felt with each change, and observations on the condition of patients removed from their natural climate to a "more favorable climate."

In the diary-type of observations, correlations between the type of weather and the subjective status of the symptoms are drawn by the authors, and conclusions are based upon such findings. It is important to note that such correlations would never be accepted as statistically significant by any statistician.

Results from the second general type of study of weather effect

This study was made possible by grants from the Arthritis and Rheumatism Foundation, the National Fund for Medical Education, and Grant A5322 of the National Institute of Arthritis and Metabolic Diseases, U. S. Department of Health, Education and Welfare.

on disease are even less convincing. Observers in a sanitarium or spa, each having an especially favorable climate to extoll, have shown by multiple case histories how much the arthritis, or asthma, or hypertension, or whatever, improved when the patients were sent to the new environment.

Little is mentioned of the other variables included in this change, such as freedom from family responsibilities and financial worries, emotional factors, decreased exertion and increased rest, supervised intake of food and liquids, etc., which obviously affect the disease state as much or more than the climate itself. Psychometric evaluations, rather than psychrometric variations, may account for the differences.

When the individual factors believed responsible for climatic effects are considered, the intellectual environment becomes dense with the "smog" of conflicting reports in a myriad of observations. It was noted in one report that barometric pressure influenced the symptoms of arthritic patients more than variations in temperature or humidity, but, in the same report, it was stated that changes did not affect all patients in the same manner.[1] By the diary method it was noted that some arthritics felt better with a rising barometer, while others improved when the barometer was falling.

In an excellent review of the published record on effects of barometric variations on both animals and man, Dordick[2] concludes: "There is no clear-cut evidence showing that variations of atmospheric pressure such as those associated with atmospheric 'lows' and

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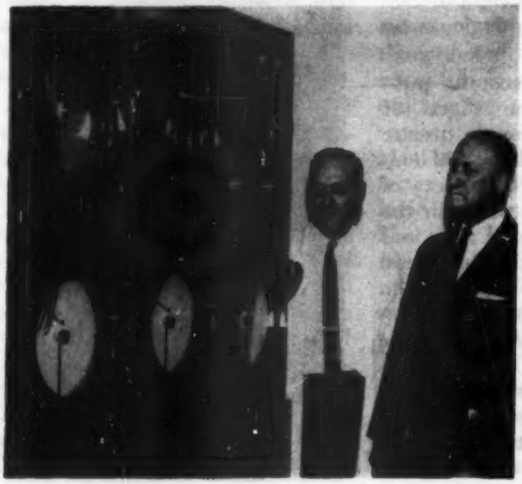

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Chilling has been condemned for centuries as the factor which is primarily responsible for worsening pain in arthritis,[3] yet many who write about the evil effects of chilling mention dampness or humidity concomitantly. It is well known, however, that chilling can be produced by decreasing temperature, increasing rate of air movement (draft), decreasing the humidity, or any combination of such factors. Even with extremes of heat and high humidity, chilling can be produced by an electric fan trained on the lightly clad body of an individual who is perspiring freely. Thus, chilling cannot be considered a single factor in weather control.

Much of the recently published information on effects of climatic factors on human disease has focused on the degree of ionization of the air. The variety of methods of producing ions in the air, and in the effects of such ions on human beings, both healthy and diseased, was well-illustrated in many conflicting reports at a recent symposium on Air Ionization held in Philadelphia. Some reports supported the findings of Kornblueh[4] that negative ions in the air in heavy concentrations had a clinically beneficial effect on the respiratory passages of patients with asthma and hay fever. Such patients had been exposed to the negative ionization knowingly, and for definite "treatment periods." These results were controlled by similar exposure periods without the ionizer turned on, but the psychological effect of any treatment on the patient was not controlled. It was also not determined by pollen counts or dust determinations whether the ionization merely acted as an electrostatic precipitator of air pollution.

The foregoing criticism of research into effects of climatic factors on human disease is not

Fig. 2 Nurse assists patient through submarine door into airlock of chamber. Note pressure valve in center of door

intended to be derogatory or scathing, but is intended to point out the many pitfalls in this type of research and show how necessary it is to have facilities for complete control of all possible psychrometric factors at all times, and to have the patients completely "blind" as far as all variations are concerned, i.e., continuously confined in a room without visible evidence of apparatus or changing conditions.

Although we have been enabled, at long last, to control temperature, humidity, barometric pressure, rate of air movement and ionization in the Controlled Climate Chamber, or "Climatron," we still have not been able to solve the problem of complete objectivity in evaluation of effects on the patients themselves. This unusual facility will be described in some detail in this presentation, and our results so far will be interpreted with the knowledge that lack of accurate measuring devices for calculation of effects may distort the true facts.

There is really nothing new about a controlled climate chamber, many of which were in use long before ours was conceived. Chambers for study of climatic effects on plants and animals are numerous, and many for study of extremes of climate on man have been in use for years, both in the research branches of the armed services and in such places as the Harvard Fatigue Laboratory and Dr. Brouha's chamber at the Haskell Laboratories of the DuPont Company.

Even the idea of making a controlled climate chamber suitable for long-term occupancy by patients is not new, since Dr. Gunnar Edstrom of the University of Lund, in Sweden, used such a facility beginning nearly 20 years ago.

What is new about our facility, however, is that we have a completely controlled chamber, for constant occupancy by two patients, for either constant or variable conditions of temperature, humidity, barometric pressure, rate of air movement or ionization, which is placed in a hospital for the exclusive study of climatic effects on disease states. Edstrom's chamber controlled only temperature and humidity, and kept them completely constant. None of the others in which all factors are controlled was constructed for comfortable continuous patient occupancy over weeks or months.

Having become convinced, over the years, that there must be something definite about the "weather effect" on disease, constructing and utilizing a completely controlled climate chamber seemed like the ideal means for determining which factor, or combination of meteorologic parameters, is responsible. The mere dream of such a chamber was far from reality. First, a sum of $125,000 had to be raised and an engineer with ingenuity had to be found. They were.

The National Institutes of Health were unable to support the project when applied to in 1953, which delayed the project for a few years. As a physician, how-

ever, I was able to interest a number of patients, relatives and friends in the idea and the money finally was available when the National Institutes of Health granted $30,000 to supplement what had been gleaned from private sources.

Finding the ingenious engineer who could design and build the facility was the greatest stroke of good luck. Charles S. Leopold, whom most of you undoubtedly remember, worked in Philadelphia. He was immediately interested and helpful, and assigned another man from his staff to the project. This was current ASHRAE President John Everetts, Jr., who has been what I facetiously term my "partner in clime" since 1953.

This man has proved to me that he is not just an engineer, but an "ingeneer," if I may be permitted to coin a word designating one who has extreme ingenuity and resourcefulness. Since I am informed that he is known to this group, perhaps he will tell in discussion some of the real engineering problems and obstacles he overcame. I just told him what I wanted; he produced the results, no matter that they might have seemed impossible.

The Controlled Climate Chamber, situated in the University of Pennsylvania Hospital, consists of a room for living and sleeping for two patients, a connecting bathroom and an airlock vestibule for entrance and exit. Walls, floor and ceiling are constructed of reinforced concrete. The window is completely sealed and consists of two layers of heavy plate glass. The room is comfortably large (15

Fig. 3 Inside the chamber, Harold W. Schaefer (center) and Lloyd Staebler explain workings of ion counter to author

x 15 ft), decorated attractively as a hotel "studio twin" might be, but with a mural wall and a large mirror to avoid any feeling of claustrophobia. The air is pumped into the chamber through multiple perforations in the false ceiling and exhausted through ducts near the floor.

The bathroom appears, superficially, like any other, but, because of atmospheric pressure differences with the external enviroment often maintained in the Chamber, some new devices in drainage were necessitated. The construction and design of the sump tank, interior vent back to the room via activated charcoal filter, and automatic valves and ejector pumps is beyond me – but it "works." I don't know which was more difficult, designing this or getting the approval to build it from the Philadelphia Department of Sanitation, who could not understand why it was necessary and why Mr. Everetts insisted on the unheard-of use of valves in the soil lines.

The airlock vestibule also called for considerable ingenuity. Air valves were inserted in submarine doors to use in equalizing pressure between outside and airlock, then airlock and chamber on entry. It also took some teaching of our entire staff to convince them of the need to lock one door before the other was opened.

Automatic recording devices on the control panel outside the chamber write a continuous record of temperature, dew point temperature, barometric pressure, rate of air movement, amount of fresh air taken into the system, glycol temperature in the refrigerating units and the charge and concentration of air ions in the Climatron. Controls for programming the desired changes in temperature, humidity, barometric pressure and rate of air movement are also located in the control panels.

Ionization of the room air is generated by coronal discharge of high voltage from three small needle ionizers in the ceiling of the room and powered by the generator in the control panel. Counting of ions is accomplished by means of an Ebert ion counter in a cabinet in the center of the room, with the recorder writing the count on the kyomographic chart on the control panel. The ion generator and the ion counter kindly were loaned to us by the Philco Corporation.

Much of this remote control apparatus could more easily have been placed within the room, but I insisted that the patients must live in a room, not in a laboratory, to get a true "blind effect," i.e., eliminating as much psychologic influence as possible from presence of apparatus in the room.

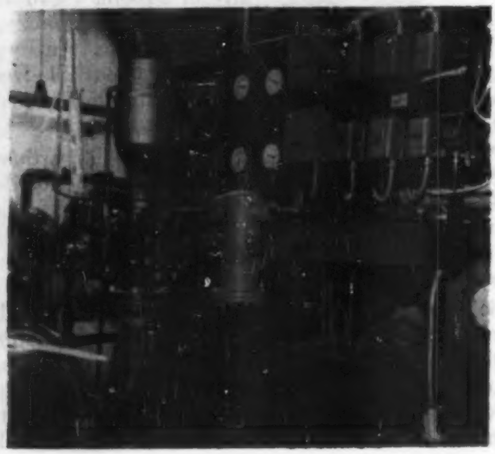
The machinery for filtering coarse and fine particles from the air, removal of odors, cooling, reheating, drying or humidifying, pressurizing and circulation of air is housed in the "engine room" below the chamber. In addition to all this apparatus and ductwork, the air compressor for the automatic controls, the plumbing connections and the sump tank and ejector pumps all vie for space in the 15 x 19-ft room. This is both

a maze and amazing for its complexity and the ability to pack so much and to do so many things in such a small space. Until I saw this room I did not appreciate how much I was taxing the ingenuity of my "ingeneer." It could compete with a submarine, an airliner or a guided missile in its total incomprehensibility to a researching physician.

Having described the apparatus, in a truly amateurish way, I can proceed to the parts of the experiment which I understand better. The apparatus permits me to keep all of the parameters of climate constant in the chamber, or to change them at predetermined rates and amounts, completely without the knowledge of patients housed in the room. The subjects are never informed of what is to be done.

Two patients at a time live continuously in the Climatron for observation periods of from 3 to 4 weeks. Meals are brought in to them, as are supplies. They are permitted to have visitors (in fact, one patient acted as baby-sitter in the Chamber for her granddaughter left with her each evening while her daughter and son-in-law attended evening classes a block away in the University). Patients are entertained by radio and television, and crafts are supervised by our occupational-therapy department. Since the incarceration makes adequate exercise difficult, the physical therapist provides instruction in exercises frequently and we bring in a mechanical ex-

Fig. 4 Interior of apparatus room, showing part of coupled intake and exhaust fans on left and refrigerating equipment (right)



erciser at times. Strange though it seems, these patients all have enjoyed their stay and many have volunteered for another period if we want them. Since they are volunteers, not in for treatment, we cannot charge them for this "vacation."

So far, ten patients with rheumatoid arthritis, four patients with pulmonary emphysema and two with osteoarthritis have been studied. All were chosen because they had been good "weather prophets" in complaining of increased discomfort before or during weather changes.

Patients were paired by sex and as closely as possible by age level, type of disease and intelligence. Partners were observed by one of our psychiatrist colleagues, who observed that they "got along" well in all cases because they exposed only about 10% of their personality to each other. Complete strangers were paired, since friends would probably have been friends no longer at the end of the incarceration.

All patients kept daily diaries, independently, noting weight four times daily, fluid intake and output, periodic mouth temperature and the time and character of any change they noted in the environment or in the symptoms or signs of the disease. At least once daily, each patient was evaluated by an experienced research physician for changes in strength, degree of joint tenderness, time taken to walk a

Fig. 5 Portion of apparatus room showing ducts, steam pipes, etc., above, ejector pumps for sump tank with controls; below, air compressor for automatic control system on right

standard distance, blood pressure, pulse and respiration. The emphysema patients had regular determinations of vital capacity and other pulmonary function tests conducted by the Section on Chest Diseases and Pulmonary Function Laboratory of our hospital. More intensive and discriminating tests will be added when indicated.

I have taken a long time getting around to reporting results for an especially good reason; they are not yet conclusive. The number of patients is still too small, the possible permutations and combinations of our parameters of study have only been sampled and this study is planned for at least a five-year period, of which we have only worked 10 months, so far. I shall not report any preliminary results in pulmonary emphysema since these are not in my province to divulge. Dr. Robert L. Mayock is in charge of this phase of study.

Preliminary results in the ten patients with rheumatoid arthritis and the two with osteoarthritis, covering 296 patient-days of observation, have shown that a constant climate in the chamber nullifies completely any effect of external weather changes as observed by the patients through the chamber window. Remarks in the diaries such as, "It's raining today, and I didn't feel it coming," have been common. All experiments started with 4 or 5 days of constant climate for baseline observations and psychological adjustment to the environment of the chamber.

Falling or rising barometric pressure, whether slow or rapid, between 28.5 and 31 in. Hg, absolute, have, so far, shown no consistent effect on arthritic symptoms or signs. Other climatic factors were kept constant during this phase of observation. There has been an interesting effect on fluid balance of the body from these pressure changes, however. Intake and output charts and weight variations seem to indicate that there may be some fluid retention with increasing pressure and increased fluid loss with falling pressures. This is not conclusive, as yet.

Positive or negative air ionization, in concentrations as high as 50,000 ions per ml of air, has produced as yet no detectable effect on arthritic symptoms or signs in any of the twelve patients. In fact, the patients were completely unaware of any environmental change with even marked variations of either atmospheric pressure or ionization.

Whereas all patients could easily detect marked variations in temperature, humidity or rate of air movement in the degree of their general comfort, none of these parameters has so far produced any consistent or significant effect on arthritic symptoms or signs when varied singly. Chilling did make some of the patients note increased stiffness and discomfort, however. Further observations are necessary on this point before conclusions can be drawn.

One of the more significant observations, so far, has been the increase in arthritic stiffness and soreness which consistently appeared the day following any ex-

ercise instruction by the physical therapist. She was instructed to visit the chamber without our knowledge some days of her choice and to omit visits on others. By diaries and examination we could detect each visit, with the attendant increase in physical activity by the patients, on the day following. It seems certain that unaccustomed physical activity, with weather factors constant, invariably increases arthritic discomfort. We wish we could find a factor or combination of climatic factors which would as consistently reproduce this worsening of symptoms, so we could study in more detail the "weather effect" on human disease.

Thus, our experiments on twelve arthritics have shown only negative results from variation of single climatic factors. We must increase our data and include many combinations and degrees of variation of factors before we can conclusively reproduce a consistently detrimental weather combination. When this is found, if it exists at all, we can then begin to study how it affects the bodily changes responsible for increased disease activity. Further studies are in progress.

BIBLIOGRAPHY

1. Medical Climatology, C. A. Mills. Charles C. Thomas, Springfield, Ill., 1939.
2. The Influence of Atmospheric Pressure on Human Beings, I. Dordick, Weather, vol. XIII, pp. 359-364, Nov. 1958.
3. Climate and the Rheumatic Diseases, W. Paul Holbrook, chapter in Arthritis and Allied Conditions, Ed. J. L. Hollander, 6th edition, Lea and Febiger, Philadelphia, 1960, pp. 557-581.
4. Electric Space Charges and Human Health, I. H. Kornblueh, Bulletin of the American Meteorological Society, vol. 41, pp. 361-367, 1960.

No. 1770

Psychrometric Charts in Review

D. D. WILE
Recold Corporation
Presidential Member
Fellow
ASHRAE

Fifty years ago at the 1911 Annual Meeting of the American Society of Mechanical Engineers, Willis H. Carrier,[1] then chief engineer of Buffalo Forge Company, stated:

"A specialized engineering field has recently developed, technically known as air conditioning, or the artificial regulation of atmospheric moisture. The application of this new art to many varied industries has been demonstrated to be of greatest economic importance. When applied to the blast furnace, it has increased the net profit in the production of pig iron from $0.50 to $0.70 per ton, and in the textile mill it has increased the output from 5 to 15%, at the same time greatly improving the quality and the hygienic conditions surrounding the operative. In many other industries, such as lithographing, the manufacture of candy, bread, high explosive and photographic films, and the drying and preparing of delicate hygroscopic materials, such as macaroni and tobacco, the question of humidity is equally important. While air conditioning has never been properly applied to coal mines, the author is convinced that if this were made compulsory, the greater number of mine explosions would be prevented."

Carrier then presented a rational psychrometric formula and a psychrometric chart (Fig. 1) by which the properties and energy relations of moist air could be determined from readings of the wet bulb and dry bulb thermometers. This chart made it possible, for the first time, to solve air conditioning problems with considerable accu-

racy and with minimum calculations.

Carrier made his analysis in 1903, but two years after graduation from Cornell University, and began his confirming tests in 1904. He credited Apjohn[2] with having expounded a similar theory in 1836. Apjohn's theory, however, was in an imperfect form and he was unable to prove his assumptions. Nor was Carrier the first to publish a psychrometric chart. Grosvenor,[3] in 1908, published an elaborate chart which, however, did not establish wet bulb lines.

As might be expected, Carrier's use and dissemination of a psychrometric chart preceded, by several years, the date of actual publication. His "hydrometric chart" appeared in a catalog (Air Washer and Humidifier) of the Buffalo Forge Company copyrighted in 1906. Mr. Logan Lewis, a co-worker, recalls that Mr. Carrier produced a blueprinted chart around 1904. This paper will review the major advances in psychrometric charts since Carrier's original contribution.

At best, the psychrometric chart is a rather complex arrangement of coordinates and parameters. As a minimum, it must display six variables: dry bulb temperature, wet bulb temperature, moisture content (specific humidity), enthalpy, relative humidity and specific volume. Over the years chart makers have had divergent views on the desired arrangement of these variables with the result that charts have taken many forms, including that shown in Fig. 2. The trend, however, has generally returned to Carrier's original arrangement. A new chart recently adopted, after much study, by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, also follows this form.

A most consistent problem in the construction of psychrometric charts is the nearly parallel relation of wet bulb and enthalpy lines, making it difficult to display both sets of lines without confusion. Carrier's original paper carried an unusual solution to this problem

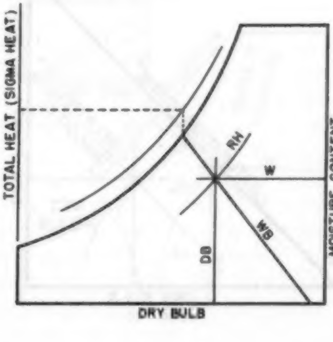


Fig. 1 Diagram of W. H. Carrier's 1911 Psychrometric Chart. This chart marked the beginning of modern air conditioning practice. Note the use of "total heat." "Enthalpy" had not come into use at that time. "Sigma heat" was a function devised by Carrier that remains constant along the wet bulb line

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What is new about our facility, however, is that we have a completely controlled chamber, for constant occupancy by two patients, for either constant or variable conditions of temperature, humidity, barometric pressure, rate of air movement or ionization, which is placed in a hospital for the exclusive study of climatic effects on disease states. Edstrom's chamber controlled only temperature and humidity, and kept them completely constant. None of the others in which all factors are controlled was constructed for comfortable continuous patient occupancy over weeks or months.

Having become convinced, over the years, that there must be something definite about the "weather effect" on disease, constructing and utilizing a completely controlled climate chamber seemed like the ideal means for determining which factor, or combination of meteorologic parameters, is responsible. The mere dream of such a chamber was far from reality. First, a sum of $125,000 had to be raised and an engineer with ingenuity had to be found. They were.

The National Institutes of Health were unable to support the project when applied to in 1953, which delayed the project for a few years. As a physician, how-

ever, I was able to interest a number of patients, relatives and friends in the idea and the money finally was available when the National Institutes of Health granted $30,-000 to supplement what had been gleaned from private sources.

Finding the ingenious engineer who could design and build the facility was the greatest stroke of good luck. Charles S. Leopold, whom most of you undoubtedly remember, worked in Philadelphia. He was immediately interested and helpful, and assigned another man from his staff to the project. This was current ASHRAE President John Everetts, Jr., who has been what I facetiously term my "partner in clime" since 1953.

This man has proved to me that he is not just an engineer, but an "ingeneer," if I may be permitted to coin a word designating one who has extreme ingenuity and resourcefulness. Since I am informed that he is known to this group, perhaps he will tell in discussion some of the real engineering problems and obstacles he overcame. I just told him what I wanted; he produced the results, no matter that they might have seemed impossible.

The Controlled Climate Chamber, situated in the University of Pennsylvania Hospital, consists of a room for living and sleeping for two patients, a connecting bathroom and an airlock vestibule for entrance and exit. Walls, floor and ceiling are constructed of reinforced concrete. The window is completely sealed and consists of two layers of heavy plate glass. The room is comfortably large (15

Fig. 3 Inside the chamber, Harold W. Schaefer (center) and Lloyd Staebler explain workings of ion counter to author

x 15 ft), decorated attractively as a hotel "studio twin" might be, but with a mural wall and a large mirror to avoid any feeling of claustrophobia. The air is pumped into the chamber through multiple perforations in the false ceiling and exhausted through ducts near the floor.

The bathroom appears, superficially, like any other, but, because of atmospheric pressure differences with the external enviroment often maintained in the Chamber, some new devices in drainage were necessitated. The construction and design of the sump tank, interior vent back to the room via activated charcoal filter, and automatic valves and ejector pumps is beyond me – but it "works." I don't know which was more difficult, designing this or getting the approval to build it from the Philadelphia Department of Sanitation, who could not understand why it was necessary and why Mr. Everetts insisted on the unheard-of use of valves in the soil lines.

The airlock vestibule also called for considerable ingenuity. Air valves were inserted in submarine doors to use in equalizing pressure between outside and airlock, then airlock and chamber on entry. It also took some teaching of our entire staff to convince them of the need to lock one door before the other was opened.

Automatic recording devices on the control panel outside the chamber write a continuous record of temperature, dew point temperature, barometric pressure, rate of air movement, amount of fresh air taken into the system, glycol temperature in the refrigerating units and the charge and concentration of air ions in the Climatron. Controls for programming the desired changes in temperature, humidity, barometric pressure and rate of air movement are also located in the control panels.

Ionization of the room air is generated by coronal discharge of high voltage from three small needle ionizers in the ceiling of the room and powered by the generator in the control panel. Counting of ions is accomplished by means of an Ebert ion counter in a cabinet in the center of the room, with the recorder writing the count on the kyomographic chart on the control panel. The ion generator and the ion counter kindly were loaned to us by the Philco Corporation.

Much of this remote control apparatus could more easily have been placed within the room, but I insisted that the patients must live in a room, not in a laboratory, to get a true "blind effect," i.e., eliminating as much psychologic influence as possible from presence of apparatus in the room.

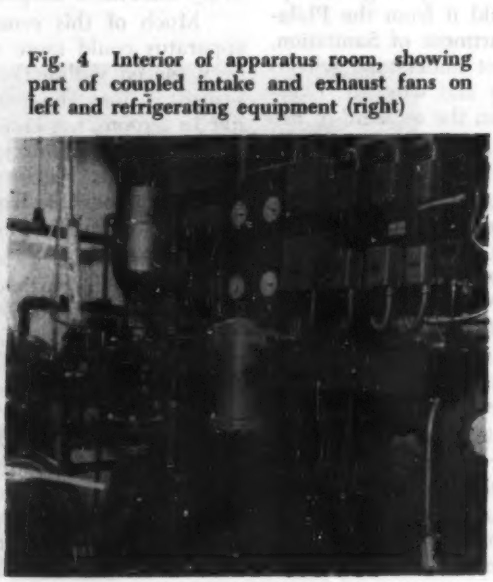
The machinery for filtering coarse and fine particles from the air, removal of odors, cooling, reheating, drying or humidifying, pressurizing and circulation of air is housed in the "engine room" below the chamber. In addition to all this apparatus and ductwork, the air compressor for the automatic controls, the plumbing connections and the sump tank and ejector pumps all vie for space in the 15 x 19-ft room. This is both

a maze and amazing for its complexity and the ability to pack so much and to do so many things in such a small space. Until I saw this room I did not appreciate how much I was taxing the ingenuity of my "ingeneer." It could compete with a submarine, an airliner or a guided missile in its total incomprehensibility to a researching physician.

Having described the apparatus, in a truly amateurish way, I can proceed to the parts of the experiment which I understand better. The apparatus permits me to keep all of the parameters of climate constant in the chamber, or to change them at predetermined rates and amounts, completely without the knowledge of patients housed in the room. The subjects are never informed of what is to be done.

Two patients at a time live continuously in the Climatron for observation periods of from 3 to 4 weeks. Meals are brought in to them, as are supplies. They are permitted to have visitors (in fact, one patient acted as baby-sitter in the Chamber for her granddaughter left with her each evening while her daughter and son-in-law attended evening classes a block away in the University). Patients are entertained by radio and television, and crafts are supervised by our occupational-therapy department. Since the incarceration makes adequate exercise difficult, the physical therapist provides instruction in exercises frequently and we bring in a mechanical ex-

Fig. 4 Interior of apparatus room, showing part of coupled intake and exhaust fans on left and refrigerating equipment (right)



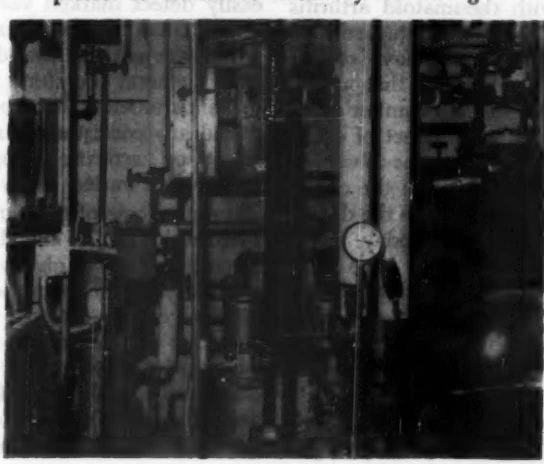
erciser at times. Strange though it seems, these patients all have enjoyed their stay and many have volunteered for another period if we want them. Since they are volunteers, not in for treatment, we cannot charge them for this "vacation."

So far, ten patients with rheumatoid arthritis, four patients with pulmonary emphysema and two with osteoarthritis have been studied. All were chosen because they had been good "weather prophets" in complaining of increased discomfort before or during weather changes.

Patients were paired by sex and as closely as possible by age level, type of disease and intelligence. Partners were observed by one of our psychiatrist colleagues, who observed that they "got along" well in all cases because they exposed only about 10% of their personality to each other. Complete strangers were paired, since friends would probably have been friends no longer at the end of the incarceration.

All patients kept daily diaries, independently, noting weight four times daily, fluid intake and output, periodic mouth temperature and the time and character of any change they noted in the environment or in the symptoms or signs of the disease. At least once daily, each patient was evaluated by an experienced research physician for changes in strength, degree of joint tenderness, time taken to walk a

Fig. 5 Portion of apparatus room showing ducts, steam pipes, etc., above, ejector pumps for sump tank with controls; below, air compressor for automatic control system on right



standard distance, blood pressure, pulse and respiration. The emphysema patients had regular determinations of vital capacity and other pulmonary function tests conducted by the Section on Chest Diseases and Pulmonary Function Laboratory of our hospital. More intensive and discriminating tests will be added when indicated.

I have taken a long time getting around to reporting results for an especially good reason; they are not yet conclusive. The number of patients is still too small, the possible permutations and combinations of our parameters of study have only been sampled and this study is planned for at least a five-year period, of which we have only worked 10 months, so far. I shall not report any preliminary results in pulmonary emphysema since these are not in my province to divulge. Dr. Robert L. Mayock is in charge of this phase of study.

Preliminary results in the ten patients with rheumatoid arthritis and the two with osteoarthritis, covering 296 patient-days of observation, have shown that a constant climate in the chamber nullifies completely any effect of external weather changes as observed by the patients through the chamber window. Remarks in the diaries such as, "It's raining today, and I didn't feel it coming," have been common. All experiments started with 4 or 5 days of constant climate for baseline observations and psychological adjustment to the environment of the chamber.

Falling or rising barometric pressure, whether slow or rapid, between 28.5 and 31 in. Hg, absolute, have, so far, shown no consistent effect on arthritic symptoms or signs. Other climatic factors were kept constant during this phase of observation. There has been an interesting effect on fluid balance of the body from these pressure changes, however. Intake and output charts and weight variations seem to indicate that there may be some fluid retention with increasing pressure and increased fluid loss with falling pressures. This is not conclusive, as yet.

Positive or negative air ionization, in concentrations as high as 50,000 ions per ml of air, has produced as yet no detectable effect on arthritic symptoms or signs in any of the twelve patients. In fact, the patients were completely unaware of any environmental change with even marked variations of either atmospheric pressure or ionization.

Whereas all patients could easily detect marked variations in temperature, humidity or rate of air movement in the degree of their general comfort, none of these parameters has so far produced any consistent or significant effect on arthritic symptoms or signs when varied singly. Chilling did make some of the patients note increased stiffness and discomfort, however. Further observations are necessary on this point before conclusions can be drawn.

One of the more significant observations, so far, has been the increase in arthritic stiffness and soreness which consistently appeared the day following any ex-

ercise instruction by the physical therapist. She was instructed to visit the chamber without our knowledge some days of her choice and to omit visits on others. By diaries and examination we could detect each visit, with the attendant increase in physical activity by the patients, on the day following. It seems certain that unaccustomed physical activity, with weather factors constant, invariably increases arthritic discomfort. We wish we could find a factor or combination of climatic factors which would as consistently reproduce this worsening of symptoms, so we could study in more detail the "weather effect" on human disease.

Thus, our experiments on twelve arthritics have shown only negative results from variation of single climatic factors. We must increase our data and include many combinations and degrees of variation of factors before we can conclusively reproduce a consistently detrimental weather combination. When this is found, if it exists at all, we can then begin to study how it affects the bodily changes responsible for increased disease activity. Further studies are in progress.

BIBLIOGRAPHY

1. Medical Climatology, C. A. Mills, Charles C. Thomas, Springfield, Ill., 1939.
2. The Influence of Atmospheric Pressure on Human Beings, I. Dordick, Weather, vol. XIII, pp. 359-364, Nov. 1958.
3. Climate and the Rheumatic Diseases, W. Paul Holbrook, chapter in Arthritis and Allied Conditions, Ed. J. L. Hollander, 6th edition, Lea and Febiger, Philadelphia, 1960, pp. 557-581.
4. Electric Space Charges and Human Health, I. H. Kornblueh, Bulletin of the American Meteorological Society, vol. 41, pp. 361-367, 1960.

1770

No. 1770

Psychrometric Charts in Review

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Fifty years ago at the 1911 Annual Meeting of the American Society of Mechanical Engineers, Willis H. Carrier,[1] then chief engineer of Buffalo Forge Company, stated:

"A specialized engineering field has recently developed, technically known as air conditioning, or the artificial regulation of atmospheric moisture. The application of this new art to many varied industries has been demonstrated to be of greatest economic importance. When applied to the blast furnace, it has increased the net profit in the production of pig iron from $0.50 to $0.70 per ton, and in the textile mill it has increased the output from 5 to 15%, at the same time greatly improving the quality and the hygienic conditions surrounding the operative. In many other industries, such as lithographing, the manufacture of candy, bread, high explosive and photographic films, and the drying and preparing of delicate hygroscopic materials, such as macaroni and tobacco, the question of humidity is equally important. While air conditioning has never been properly applied to coal mines, the author is convinced that if this were made compulsory, the greater number of mine explosions would be prevented."

Carrier then presented a rational psychrometric formula and a psychrometric chart (Fig. 1) by which the properties and energy relations of moist air could be determined from readings of the wet bulb and dry bulb thermometers. This chart made it possible, for the first time, to solve air conditioning problems with considerable accu-

racy and with minimum calculations.

Carrier made his analysis in 1903, but two years after graduation from Cornell University, and began his confirming tests in 1904. He credited Apjohn[2] with having expounded a similar theory in 1836. Apjohn's theory, however, was in an imperfect form and he was unable to prove his assumptions. Nor was Carrier the first to publish a psychrometric chart. Grosvenor,[3] in 1908, published an elaborate chart which, however, did not establish wet bulb lines.

As might be expected, Carrier's use and dissemination of a psychrometric chart preceded, by several years, the date of actual publication. His "hydrometric chart" appeared in a catalog (Air Washer and Humidifier) of the Buffalo Forge Company copyrighted in 1906. Mr. Logan Lewis, a co-worker, recalls that Mr. Carrier produced a blueprinted chart around 1904. This paper will review the major advances in psychrometric charts since Carrier's original contribution.

At best, the psychrometric chart is a rather complex arrangement of coordinates and parameters. As a minimum, it must display six variables: dry bulb temperature, wet bulb temperature, moisture content (specific humidity), enthalpy, relative humidity and specific volume. Over the years chart makers have had divergent views on the desired arrangement of these variables with the result that charts have taken many forms, including that shown in Fig. 2. The trend, however, has generally returned to Carrier's original arrangement. A new chart recently adopted, after much study, by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, also follows this form.

A most consistent problem in the construction of psychrometric charts is the nearly parallel relation of wet bulb and enthalpy lines, making it difficult to display both sets of lines without confusion. Carrier's original paper carried an unusual solution to this problem

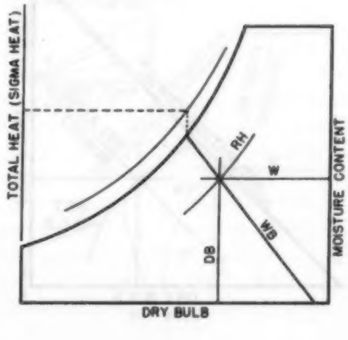

Fig. 1 Diagram of W. H. Carrier's 1911 Psychrometric Chart. This chart marked the beginning of modern air conditioning practice. Note the use of "total heat." "Enthalpy" had not come into use at that time. "Sigma heat" was a function devised by Carrier that remains constant along the wet bulb line

and since then there have been many other approaches. The situation has not been without controversy; therefore, a brief review of the underlying factors appears in order.

The wet bulb temperature was defined by Carrier as "The Temperature of Adiabatic Saturation." This process can be visualized in simple form as air moving through an insulated tube having wetted wicks or other saturating devices. As the air picks up moisture, there is an interchange of sensible and latent heat whereby the dry bulb temperature of the air is reduced. Wet bulb thermometers along the path of the air will show substantially the same readings even though the air enters dry and leaves saturated. Water in the saturator will assume the wet bulb temperature and if the process is continuous, water must be added at the wet bulb temperature. It is significant that in this process there will be an increase of enthalpy in an amount supplied by the make-up water. Thus, on a psychrometric chart, a line of constant wet bulb temperature is not a line of constant enthalpy but deviates slightly from it. Numerous expedients have been introduced to either eliminate the enthalpy lines or to orient them to a less confusing position.

Carrier used the term "Sigma Heat" to define a function that remains constant along the wet bulb line. It was the enthalpy of the moist air minus the heat of the liquid contained in it. Enthalpy was computed easily by adding the sensible heat of the liquid to the "Sigma Heat." By including a "Sigma Heat" scale (see Fig. 1), the Carrier chart thus provided enthalpy data without the necessity of showing the enthalpy lines that are so nearly parallel to the wet bulb lines. The "Sigma Heat" concept was used for many years by Carrier Corporation and others. Frequently, in making load calculations in the comfort air conditioning range, the corrections to "Sigma Heat" were ignored because the deviations tended to offset one

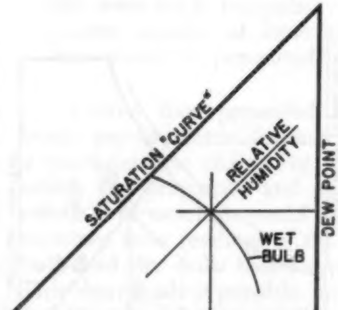

Fig. 2 Diagram of the C. A. Bulkley Chart Using Non-Uniform Scales. The various coordinates and parameters of a psychrometric chart can be placed in innumerable arrangements. The modified logarithmic scales of this chart provided more uniform accuracy over a wide range of temperature and humidity but the chart was unsuited for graphic solution of air-treating problems

another and the resulting error was generally less than 2% of the total load.

A new approach for eliminating enthalpy lines was introduced in 1934 on a copyrighted chart by F. O. Urban of General Electric Company. Urban used a diagonal enthalpy scale and, instead of a grid of enthalpy lines, he extended the wet bulb lines from the saturation curve to this scale and thus indicated "enthalpy at saturation." True enthalpy could be obtained from "enthalpy at saturation" by deducting the difference in enthalpy of the moisture contents at saturation and at the state point. In comfort air conditioning calculations, this correction usually was ignored with but slight error. Chart users preferred the "Enthalpy at Saturation" arrangement and eventually the "Sigma Heat" concept fell into disuse except for variable elevation charts, as explained later. There remained, however, a real need for a practical chart showing true enthalpy without the confusion of the nearly parallel lines. Two notable solutions will be discussed.

In 1945, Goff[4] introduced a chart with "reduced enthalpy" lines, actual enthalpy being obtained by adding one thousand times the moisture content to the "reduced enthalpy" value. This device required additional calculations (even for approximate results) and although it rotated the "reduced enthalpy" lines away from the wet bulb lines, they became somewhat parallel to the dry bulb lines.

In 1946, Palmatier[5] devised "enthalpy deviation" contours on a chart (Fig. 3) that used the "enthalpy at saturation" concept. These show at a glance the deviation of enthalpy along the wet bulb lines. The contours are widely spaced to avoid complication of the chart structure, yet permit accurate determination of true enthalpy. They also indicate the magnitude of error if the deviations are ignored and can be applied to tabular data (enthalpy at saturation) when more precise values are required. The

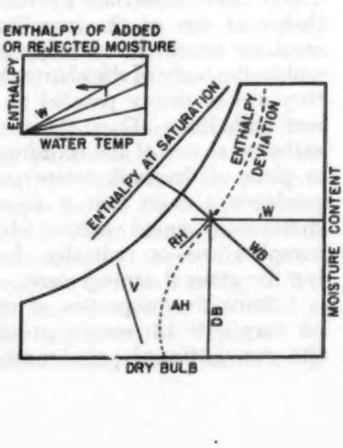

Fig. 3 Diagram of the New ASHRAE Chart. Enthalpy deviation contours, as shown dotted, permit accurate determination of enthalpy while avoiding the confusion of the enthalpy lines which are so nearly parallel to the wet bulb lines

new ASHRAE chart incorporates the enthalpy deviation concept.

Usage of psychrometric charts may be divided into two general classes. Most frequently the use is to determine certain physical properties from known measurements, such as finding relative humidity, enthalpy, specific volume and moisture content when only the wet and dry bulb temperatures are known. A more sophisticated use is to show the result of mixing unlike streams of air or to show the path, on the chart, of a stream of air as heat and moisture are added or removed. This latter usage requires, for precise results, that the chart construction be thermodynamically sound.

Carrier's original chart, and for many years those that followed, used coordinates of moisture content and dry bulb temperature. Such a chart will give accurate values for the properties of air at a given point. However, a straight line between two points does not indicate precisely the state of mixtures between the points. Fortunately, the error is negligible in the comfort air conditioning range.

In 1923, Mollier,[6,7] of steam chart fame, proposed coordinates of enthalpy and moisture content, with dry bulb as a parameter. If the enthalpy coordinate is at an oblique angle, the chart will have the conventional appearance, with the exception that the dry bulb lines are not exactly parallel with each other but diverge slightly with increasing moisture content (Fig. 4). On a chart such as the new ASHRAE chart covering the

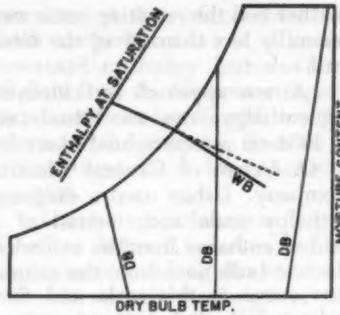

Fig. 4 Dry Bulb Lines Diverge on a Chart That Is Thermodynamically Sound. Modern practice is to use coordinates of moisture content (horizontal) and enthalpy (at an oblique angle). The divergence of the dry bulb lines, here shown exaggerated, is hardly detectable to the eye on charts for the usual air conditioning range

usual air conditioning temperature range the divergence of the dry bulb lines is hardly detectable to the eye.

A chart constructed with enthalpy as one of the coordinates need not retain the enthalpy lines within the body of the chart where they are so nearly parallel to the wet bulb lines. Thus, the use of enthalpy as one of the coordinates, in place of dry bulb temperature, produces a chart that is thermodynamically sound without adding complications or radically changing the general arrangement.

Since the properties of moist air vary with barometric pressure, the conventional psychrometric

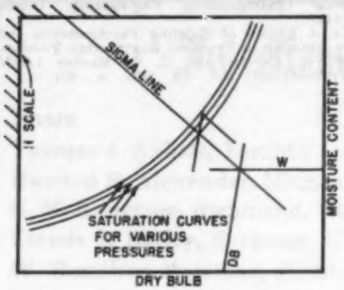

Fig. 5 The Variable Elevation Chart. Charts with multiple saturation curves cover a wide range of elevations but are inconvenient to use. Wet bulb lines take on different values for each elevation, their slope being determined by Carrier's "sigma heat" concept. For day-to-day use at a specific elevation, it is generally desirable to construct a chart for that elevation

chart (usually constructed for sea-level barometric pressure) will give erroneous results at other elevations.[8] In an ordinary air conditioning calculation the load may be in error by 10% if the elevation is 4000 ft different from the chart basis.

Variable elevation charts have been published by numerous authors,[9,10,11] including an excellent example by the Bureau of Mines[12] in 1947. These charts have a family of saturation curves for various elevations, as shown in Fig. 5. Wet bulb lines cannot be identified as such but must be replaced by adiabatic saturation (Carrier's Sigma Function) lines. These lines take on the wet bulb temperature indicated by the intersection of dry bulb lines with the saturation curve for any specific elevation.

The variable elevation chart is excellent for occasional reference but is inconvenient for day-to-day use at a specific elevation. It is more convenient to construct a chart for such use from data that are readily available.[4]

Carrier's historic paper, with proof of his assumptions, along with his practical psychrometric chart and psychrometric data, laid a sound foundation for an industry that has grown far beyond his optimistic predictions. As a pioneer in a new industry, he not only promoted its progress with great zeal but he unselfishly made his knowledge available to others. He truly deserved the title "The Father of Air Conditioning."

REFERENCES

1. Rational Psychrometric Formulae, W. H. Carrier (ASME Transactions, Vol. 33, 1911, p. 1005).
2. James Apjohn (Irish Academy Transactions, Vol. 17, 1837, p. 275).
3. Calculations for Dryer Design, W. M. Grosvenor (AIChE Transactions, Vol. 1, 1908, p. 184).
4. American Society of Heating and Ventilating Engineers GUIDE, Vol. 23, 1945.
5. A New Psychrometric Chart, E. P. Palmatier and D. D. Wile (Refrigerating Engineering, Vol. 54, July 1946).
6. Ein Neves Diagram fur Dampfluftgemische, R. Mollier (Zeitschrift VD I, Vol. 69, September 1923, p. 869).
7. Das ix Diagramm fur Dampfluftgemische (Zeitschrift VD I, Vol. 23, 1929, p. 1009).
8. Psychrometric Charts, Past & Present, D. D. Wile (ASHRAE JOURNAL, August 1959).
9. The Psychrometric Chart, Its Application and Theory, W. Goodman (Heating, Piping and Air Conditioning, Vols. 11 and 12, June 1939 through August 1940).
10. Heat and Vapor Transfer of Cooling Coils at High Elevations, N. Sharpe (Refrigerating Engineering, Vol. 6, p. 1055).
11. Psychrometric Charts for Use at Altitudes Above Sea Level, B. H. Jennings and A.

Tortoni (Refrigerating Engineering, June 1954, p. 71).
12. A Mine Air Conditioning Chart, G. E. McElroy (Bureau of Mines Publication R. I. 4165, December 1947).
13. Psychrometry in the Frost Zone, D. D. Wile (Refrigerating Engineering, October 1944).
14. A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems, W. H. Carrier and C. O. Mackey (ASME Transactions, Vol. 59, 1937, p. 33).

In Memoriam 1961

NAME	JOINED
Thomas J. Abbott, Toronto, Ont.	1938
Howard B. Alexander, Minneapolis, Minn.	1946
G. M. Anderson, Richmond, Va.	1941
Claude L. Bailey, Syracuse, N. Y.	1955
W. Hamilton Bateman, Pittsburgh, Pa.	1944
Raymond E. Bennis, Old Westbury, N. Y.	1932
Clarence D. Berkeley, Richland, Wash.	1944
Vivian F. Berton, Ottawa, Ont.	1956
O. K. Bjorklund, Buenos Aires, Argentina	1939
Harold A. Brangs, Newark, N. J.	1952
Benjamin A. Broom, (*Life Member*), Marshville, N. C.	1914
William Hugh Calahan, Fort Thomas, Ky.	1954
Byron L. Casey, (*Life Member*), Chicago, Ill.	1921
F. D. Chinoy, Bombay, India	1957
John B. Coleman, (*Life Member*), Providence, R. I.	1920
John W. Collins, Sundridge, Ont.	1946
Richard F. Connell, (*Life Member*), Detroit, Mich.	1916
A. Corbman, Philadelphia, Pa.	1950
R. A. Crawford, St. Clair Shores, Mich.	1945
Henri Dagenais, Montreal, Que.	1950
James F. Dailey, Tampa, Fla.	1944
Fred C. Dawson, Canton, Mass.	1944
John E. Day, Sacramento, Calif.	1952
William B. Dias, New York, N. Y.	1948
Alfred M. Dion, Toronto, Ont.	1948
Stanley Dreskin, New York, N. Y.	1949
Charles L. Dugan, Tampa, Fla.	1956
E. O. Eastwood, (*Presidential Member*), Seattle, Wash.	1921
Edwin C. Evans, (*Life Member*), N. El Monte, Calif.	1919
S. B. Evans, W. Vancouver, British Columbia	1952
Charles F. Eveleth, (*Life Member*), Concord, Mass.	1911
A. H. Foster, Vancouver, British Columbia	1957
Lloyd L. Garrett, St. Petersburg, Fla.	1960

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Name	Joined
Joseph T. Gier, Los Angeles, Calif.	1954
Louis Gold, Brooklyn, N. Y.	1958
Elmo Hall, Englewood, Colo.	1944
G. Ross Hamilton, Ottawa, Ont.	1952
Lloyd F. Hannan, San Francisco, Calif.	1945
James W. Hardy, Park Ridge, Ill.	1948
James E. Hart, Norfolk, Va.	1947
E. P. Heckel, *(Life Member)*, Chicago, Ill.	1918
Mahlon L. Heist, Philadelphia, Pa.	1957
Joseph G. Heitman, Midlothian, Ill.	1946
Arthur F. Hinrichsen, *(Life Member)*, New York, N. Y.	1928
A. Israel, Rockville Center, N. Y.	1945
Frederick M. Jones, Minneapolis, Minn.	1942
Sewell O. Jordin, Seattle, Wash.	1958
Jeff Kearn, Windsor, Ont.	1943
Walter E. Keist, Pittsburgh, Pa.	1944
Parke H. Kinietz, Galesburg, Ill.	1951
J. Donald Kreble, Bellwood, Ill.	1958
H. F. Kruger, Troy, N. Y.	1941
James H. Legg, New York, N. Y.	1953
G. L. Lehle, *(Life Member)*, Chicago, Ill.	1918
Edward D. Martens, Hempstead, N. Y.	1937
Trygve Maseng, Louisville, Ky.	1956
Dale G. McCauley, Murfreesboro, Tenn.	1946
Henry G. McCullough, Philadelphia, Pa.	1944
Melvin H. McWilliams, Cincinnati, Ohio	1953
Floyd E. Merryman, Oklahoma City, Okla.	1955
Theodore I. Messer, Phoenix, Ariz.	1961
William O. Miller, Oakland, Calif.	1947
Robert K. Milward, *(Life Member)*, Detroit, Mich.	1943
Herbert S. Moore, *(Life Member)*, Toronto, Ont.	1923
G. L. Moore, *(Life Member)*, Chicago, Ill.	1923
R. Emmet Moylan, Los Angeles, Calif.	1955

In Memoriam 1961 *(Continued)*

NAME	JOINED
Bertram M. Natkin, White Plains, N. Y.	1951
Charles F. Newport, *(Life Member)*, Flossmoor, Ill.	1906
Edward Patterson, Kenmore, N. Y.	1947
Leonard Peller, Philadelphia, Pa.	1934
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Clarence Quall, Clintonville, Wis.	1956
Hans F. W. Rasmussen, *(Life Member)*, New Orleans, La.	1944
George J. Redmond, Kansas City, Mo.	1950
Eric G. Rowledge, Cowley, Oxford, England	1937
Reid Sanders, S. Boston, Mass.	1948
Leon L. Sawyer, Albuquerque, N. M.	1958
Cleve A. Sewell, Colonia, N. J.	1946
John H. Shrock, *(Life Member)*, LaPorte, Ind.	1924
Kornelius Siemens, Edmonton, Alta.	1955
D. Kennard Smith, Fernwood, Pa.	1947
Elbridge D. Smith, Tucson, Ariz.	1959
Elmer G. Smith, College Station, Tex.	1929
Stanley C. Smith, Ottawa, Ont.	1953
John H. Spence, St. Louis, Mo.	1942
Cyril R. Squance, Barrie, Ont.	1947
John M. Staveley, Norwood, Mass.	1956
Kalman Steiner, Baltimore, Md.	1949
Harry L. Stevens, Hutchinson, Kans.	1924
A. B. Stickney, *(Presidential Member)*, Flushing, N. Y.	1932
William R. Stockwell, *(Life Member)*, Michigan City, Ind.	1901
A. G. Sutcliffe, *(Life Member)*, Park Ridge, Ill.	1922
R. H. Sweeney, Minneapolis, Minn.	1939
Adolph G. Syska, Bronxville, N. Y.	1933
Thomas G. Thurston, *(Life Member)*, Elmwood Park, Ill.	1921
Francis M. Tompkins, Washington, D. C.	1944
J. G. Trawick, Birmingham, Ala.	1958

In Memoriam 1961 *(Continued)*

Name	Joined
R. E. Vernon, Jackson, Miss.	**1953**
Robert J. Waalkes, Holland, Mich.	**1953**
Jack H. Waggoner, Newark, Ohio	**1937**
William J. Way II, Houston, Tex.	**1941**
Wendell Westover, Albany, N. Y.	**1936**
William H. Wheeler, New York, N. Y.	**1948**
Frank C. Winterer, (*Life Member*), St. Paul, Minn.	**1920**
David D. Zink, Tulsa, Okla.	**1950**

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